TRANSACTIONS

OF

THE SOCIETY OF NAVAL ARCHITECTS AND MARINE ENGINEERS

1920

VOL. XXVIII 1920

TRANSACTIONS

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OF

THE SOCIETY OF NAVAL ARCHITECTS AND MARINE ENGINEERS

EDITED BY THE SECRETARY

PRINCIPAL OFFICE OF THE SOCIETY
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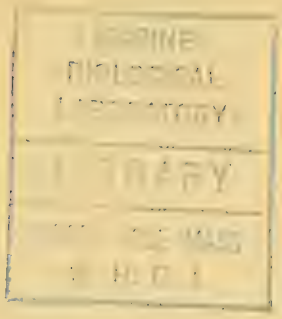

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OFFICERS OF THE SOCIETY

President.

WASHINGTON L. CAPPS.

Past Presidents and Life Members.

FRANCIS T. BOWLES, STEVENSON TAYLOR.

Past President and Life Associate Member.

ROBERT M. THOMPSON.

Honorary Vice-Presidents.

F. L. FERNALD, W. M. McFARLAND,
F. E. KIRBY, W. J. BAXTER,
D. W. TAYLOR, W. F. DURAND.

Vice-Presidents.

Term expires December 31, 1921:

LEWIS NIXON, C. H. PEABODY,
H. I. CONE, J. W. POWELL.

Term expires December 31, 1922:

A. P. NIBLACK, H. D. GOULDER,
R. M. WATT, C. P. WETHERBEE.

Term expires December 31, 1923:

H. L. FERGUSON, F. L. DuBOSQUE,
H. A. MAGOUN, W. A. DOBSON.

Members of Council.

Term expires December 31, 1921:

W. D. FORBES, R. H. M. ROBINSON, J. H. MULL,
ANDREW FLETCHER, W. G. COXE, ROBERT HAIG.

Term expires December 31, 1922:

J. H. LINNARD, W. L. R. EMMET, W. J. DAVIDSON,
C. A. McALLISTER, J. H. GARDNER, H. P. FREAR.

Term expires December 31, 1923:

H. C. SADLER, C. F. BAILEY, E. H. RIGG,
D. H. COX, W. H. TODD, WM. McENTEE.

Associate Members of Council.

Term expires December 31, 1921:

E. M. BULL, C. M. WALES.

Term expires December 31, 1922:

R. A. C. SMITH, A. W. GOODRICH.

Term expires December 31, 1923:

H. L. ALDRICH, H. H. RAYMOND.

Executive Committee.

WASHINGTON L. CAPPS, *Ex-Officio.*

STEVENSON TAYLOR, W. M. McFARLAND, J. W. POWELL,
ANDREW FLETCHER, F. L. DuBOSQUE, D. H. COX, *Ex-Officio.*

Secretary-Treasurer.

DANIEL H. COX.

ARTICLES OF INCORPORATION

THE SOCIETY OF NAVAL ARCHITECTS AND MARINE ENGINEERS

State of New York,
City and County of New York. }

We, the subscribers, Wm. H. Webb, Chas. H. Cramp, George E. Weed, H. Taylor Gause, William T. Sampson, Horace See, Frank L. Fernald, Francis T. Bowles, Washington L. Capps, Edwin D. Morgan, George W. Quintard, Harrington Putnam and Jacob W. Miller, being persons of full age and citizens of the United States, of whom a majority—namely, William H. Webb, George E. Weed, Horace See, Edwin D. Morgan, George W. Quintard, Harrington Putnam, Frank L. Fernald, and Jacob W. Miller are citizens of and residents of and within this State, desiring to associate ourselves for scientific purposes under, and pursuant to, an Act of the State of New York providing for the incorporation of benevolent, charitable, scientific, and missionary societies, passed April 12, 1848, and the several acts amending or supplementing the same, do hereby, in accordance with the requirements thereof, certify as follows:—

First. The name of title by which the Society shall be known in law is The Society of Naval Architects and Marine Engineers.

Second. The particular business and objects of such Society are the promotion of practical and scientific knowledge in the arts of shipbuilding and marine engineering and the allied professions, and in furtherance of this object, to hold meetings for social intercourse among its members, and the reading and discussion of professional papers, and to circulate by means of publication the knowledge thus obtained.

Third. The number of directors, trustees, or managers to manage the Society shall be seven, and shall consist of a President, a Secretary, and five Members of Council.

Fourth. The names of the trustees, directors, or managers of the Society for the first year of its existence are:—President, Clement A. Griscom; Secretary, Washington L. Capps; Members of Council, Francis T. Bowles, H. Taylor Gause, Chas. H. Loring, Lewis Nixon, Harrington Putnam.

Fifth. The business of the Society is to be conducted, and its place of business and principal office is to be located, in the City and County of New York.

In Witness Whereof we have made, signed, and acknowledged this Certificate, this 28th day of April, 1893.

WILLIAM H. WEBB.
CHAS. H. CRAMP.
H. T. GAUSE.
GEORGE E. WEED.
W. T. SAMPSON.
HORACE SEE.
F. L. FERNALD.
FRANCIS T. BOWLES.
W. L. CAPPS.
E. D. MORGAN.
GEORGE W. QUINTARD.
HARRINGTON PUTNAM.
J. W. MILLER.

CITY AND COUNTY OF NEW YORK, *ss:*

On this 28th day of April, 1893, before me personally appeared William H. Webb, Charles H. Cramp, H. Taylor Gause, George E. Weed, William T. Sampson, Horace See, Frank L. Fernald, Francis T. Bowles, Washington L. Capps, and Edwin D. Morgan, to me known and known to me to be the persons described in and who executed the foregoing certificate, and severally acknowledged to me that they executed the same.

JAMES FORRESTER,
Notary Public, Kings Co., Cert. N. Y. Co.

CITY AND COUNTY OF NEW YORK, *ss:*

On this 1st day of May, 1893, before me personally appeared George W. Quintard and Harrington Putnam, to me known and known to me to be the individuals described in and who executed the foregoing certificate, and they severally acknowledged to me that they executed the same.

JAMES FORRESTER,
Notary Public, Kings Co., Cert. N. Y. Co.

CITY AND COUNTY OF NEW YORK, *ss:*

On this 9th day of May, 1893, before me personally appeared Jacob W. Miller, to me known and known to me to be one of the individuals described in and who executed the foregoing certificate, and he duly acknowledged to me that he executed the same.

JAMES FORRESTER,
Notary Public, Kings Co., Cert. N. Y. Co.

(ENDORSED.)

Upon reading the within Certificate for the Incorporation of the Society of Naval Architects and Marine Engineers, I hereby approve and consent to the incorporation thereof and the within Certificate and filing thereof, and direct that the same be filed in the office of the Clerk of the City and County of New York.

Dated New York, May 10, 1893.

EDWD. PATTERSON,
Justice of the Supreme Court in the State of New York in and for the City and County of New York.

CONSTITUTION AND BY-LAWS OF THE SOCIETY OF NAVAL ARCHITECTS AND MARINE ENGINEERS

Article I.

Name and Object.

1. The name of the Association shall be "The Society of Naval Architects and Marine Engineers."

2. Its object shall be the promotion of the art of ship building,commercial and naval.

3. In furtherance of this object, annual meetings shall be held for the reading and discussion of appropriate papers and interchange of professional ideas, thus making it possible to combine the results of experience and research on the part of shipbuilders, marine engineers, naval officers, yachtsmen, and those skilled in producing the material from which ships are built and equipped.

Article II.

Membership.

1. The Society shall consist of Members, Associates, Juniors, Honorary Members and Honorary Associates.

2. *Members.*—(1) The class of Members shall consist exclusively of Naval Architects, Marine and Mechanical Engineers, including Professors of Naval Architecture or Mechanical Engineering in colleges of established reputation.

(2) A Candidate for this class must not be less than twenty-eight years of age and comply with the following regulations: He shall submit to the Council a statement showing that he has been engaged in the practice of his profession, in a responsible capacity, for at least five years, and setting forth the grounds upon which he bases his claim to membership. This statement shall be signed by three Members, who shall certify to their personal knowledge of the candidate and approval of his statement.

(3) In the case of persons not American citizens, the signatures of three Members shall be required in confirmation of their personal knowledge of the candidate's scientific attainments.

(4) If three-fourths of the members of the Council present are in favor of the admission of the candidate his name shall be submitted to the Members of the Society at the next meeting; the voting to be by ballot, should a ballot be demanded.

3. *Associates.*—(1) The class of Associates shall consist of persons, who, by profession, occupation, or scientific attainments, are qualified to discuss the qualities of a ship.

(2) Candidates for this class shall submit to the Council a written statement of their qualifications for membership; this shall be signed by two Members or Associates who shall certify to their personal knowledge of the candidate and approval of his statement. If considered by three-fourths of the Council present duly qualified for associate membership their names shall be submitted to the Society at the next meeting, to be

voted upon by the Members and Associates; the voting to be by ballot, should a ballot be demanded.

4. The proportion of favorable votes for deciding the election of Members and Associates shall be at least four-fifths of the number recorded.

5. *Juniors.*—(1) The class of Juniors shall consist of graduates of technical schools of established reputation, or persons who have had not less than two years' practical experience in marine engine works or shipyards.

(2) Candidates must be at least eighteen years of age and certify their intention to continue in the profession and become naval architects or marine engineers.

(3) Juniors shall be eligible for transfer to the class of Members or Associates after fulfilling the necessary conditions; when they are twenty-eight years of age they shall be offered the option of being transferred to the class of Members or Associates in conformity with their qualifications; but if they do not accept such offer or do not qualify, they shall be dropped from the rolls of the Society.

(4) The admission of Juniors shall be by a favorable vote of three-fourths of the members of the Council present.

(5) Juniors shall have no voice in the government of the Society or admission of members.

6. *Honorary Members and Honorary Associates.*—The Council may elect Honorary Members and Honorary Associates, the total number not to exceed twenty-five. They shall be persons of acknowledged eminence in their profession upon whom the Council may see fit to confer an honorary distinction.

7. *Benefactors and Permanent Members.*—The Council may establish a list of Benefactors. It may also elect as Permanent Members of the Society such Members or Associate Members as shall, in the opinion of the Council, by reason of notably liberal contributions to the Society, merit special recognition. Permanent Members shall, from the date of their election, be relieved from annual dues, and shall have the right, subject to the approval of Council, to designate, by will or otherwise, their successors. The conditions to be met before placing any name on the list of Benefactors and the regulations governing the election of Permanent Members shall be prescribed by the Council.

Article III.

Dues, Suspension and Expulsion of Members.

1. The entrance fees, payable on admission to the Society, shall be as follows:

Members and Associates, fifteen dollars; Juniors, ten dollars; Honorary Members and Honorary Associates, no fees.

2. The annual dues shall be as follows:

Members and Associates, fifteen dollars; Juniors, ten dollars; Honorary Members and Honorary Associates, no dues.

3. A member transferred from one grade to another shall pay the difference between the entrance fees of the two grades, and his annual dues shall be those of the grade to which transferred.

4. The annual dues shall be payable in advance on the first day of January. The Secretary shall notify each member of the amount due for the ensuing year at the time of giving notice for the annual meeting. On notification of his election each member

shall pay his entrance fee. If he desires to receive the published transactions of the Society for the meeting at which he is elected, he shall pay ten dollars in addition to the entrance fee.

5. Members and Associates can compound for all future dues and become Life Members or Life Associates by making a single payment of three hundred dollars and signing an agreement to conform to any future amendments to the Constitution and By-Laws.

6. Members are entitled to no return of fees upon severing their connection with the Society.

7. Any member whose dues are more than three months in arrears shall be notified by the Secretary. Should his dues become six months in arrears he shall be again notified by the Secretary and his rights as a member suspended. Should his dues become one year in arrears the delinquent member shall forfeit his membership in the Society unless the Council may deem it expedient to extend the time of payment.

8. The Council may, in its discretion, temporarily suspend the annual payment of dues by any member whose circumstances have become such as to make such payment impossible, and may, under similar circumstances, remit the whole or part of dues in arrears.

9. No name shall be entered on the rolls as Member, Associate or Junior, nor shall the privileges of membership be enjoyed until the payments required in paragraphs 1 to 4 of this article have been made; if the payment be delayed for more than six months from the date of election, the same shall be void unless the Council otherwise direct.

10. (1) Should the expulsion of any member be judged expedient by five or more members, they must draw up and sign a proposal requesting such expulsion, delivering the same to the Secretary, to be laid before the Council.

(2) If the Council do not find reason to concur in the proposal, no entry thereof shall appear in the minutes, nor shall any public discussion thereon be permitted.

(3) If, however, the Council find the charges contained in the proposal for expulsion substantiated, the accused member shall be notified and given an opportunity to resign. If he avails himself of this privilege, no entry shall be made on the minutes nor public discussion of the case be permitted. But if he declines to resign and offers no satisfactory explanation of the charge, the whole case shall be submitted to a special meeting of the Society.

(4) If two-thirds of the members at this special meeting (providing there be not less than twenty present) vote for expulsion, the Chairman of the meeting shall cause the accused to be expelled from the Society and direct the Secretary to notify the accused of this action.

Article IV.

Officers.

1. The officers of the Society shall consist of a President, Past Presidents, Honorary Vice-Presidents, twelve Vice-Presidents, twenty-four Members of Council, and a Secretary and Treasurer.

2. Both Members and Associates are eligible for the offices of President, Honorary Vice-President, Vice-President and Member of Council, but three-fourths of the Council shall be Members.

3. Prior to the date of the annual meeting of the Society of the year with which the term of the President expires, the Council shall nominate a candidate for the office of President, whose name shall be presented to the Society for election at the annual meeting. Any other candidate whose nomination, signed by at least sixty Members and Associates, shall have been submitted to the Secretary prior to the annual meeting shall also be presented. The candidate receiving the highest number of votes shall be the President for the ensuing three years. The first President under this rule was elected at the annual meeting in 1906 and his term of office began January 1st, 1907. The President shall not be eligible for election as his own successor.

4. (1) The term of office of the Vice-Presidents shall be three years. The terms of the Vice-Presidents which, under a previous provision of the Constitution, should have expired in 1911 and 1912 are regarded as having expired on December 31st, 1911. Those terms which, under the same conditions, should have expired in 1913 and 1914 are regarded as having expired on December 31st, 1912; those which should have expired in 1915 and 1916 are regarded as having expired on December 31st, 1913. Beginning with the four Vice-Presidents elected for the term ending December 31, 1914, there shall be four Vice-Presidents elected each year to take the place of those whose terms expire. The Vice-Presidents to fill the vacancies occurring each year in any class shall be elected by the Council from their own membership. Retiring Vice-Presidents shall be eligible for re-election.

(2) Honorary Vice-Presidents shall be chosen from the list of Vice-Presidents who have had at least ten years' service as Vice-President. They shall be chosen at the meeting of the Council next prior to the annual general meeting of the Society and must be the unanimous choice of all members of Council present. Not more than two Vice-Presidents may be elected Honorary Vice-Presidents in any one year.

5. (1) The term of office of the Members and Associate Members of Council shall be three years. Prior to August 1st of each year the Secretary shall mail to each Member and Associate a list of the names of the Members and Associate Members of Council whose terms expire with the current year, and shall enclose a blank on which each Member may, if he so desires, nominate one additional candidate for Member of Council and one additional candidate for Associate Member of Council; and each Associate Member may, if he so desires, nominate one additional candidate for Associate Member of Council. No nominations received by the Secretary after August 31st shall be considered.

(2) Prior to September 1st of each year the President shall appoint a nominating committee of five, at least three of whom shall be Members. This committee shall scrutinize the nomination slips sent in by Members and Associates and prepare a ballot. This ballot shall contain the names of all the retiring members and Associate Members of Council who shall not have declined re-election and a sufficient number of additional names to make a total of nine candidates for Member of Council and three candidates for Associate Member of Council. The names listed on the ballot, other than those of retiring Members and Associate Members of Council, shall be chosen by the nominating committee from those who have received the highest number of votes on the nominating lists returned by members of the Society.

(3) Should the number of names which have received at least ten votes in the case of nominees for Members of Council and five votes in the case of nominees for Associate Members of Council be insufficient to complete the number on the ballot list above provided for in sub-paragraph (2), the nominating committee shall add enough names

to bring the total on the ballot up to nine candidates for Member and three candidates for Associate Member of Council.

(4) The ballots shall be sent by mail as soon as possible after September 1st of each year to all Members and Associate Members of the Society. Each Member may vote for not more than six Members of Council and two Associate Members of Council, and each Associate may vote for not more than two Associate Members of Council. The ballots shall be returned by mail to the Secretary and canvassed by a committee of three members appointed by the President. This canvassing committee shall report the results of the election to the Council at its meeting immediately prior to the Annual Meeting of the Society.

(5) When the ballots have been sent out by the Secretary, should there be a desire on the part of Members or Associates to suggest another list of candidates for the Council, any twenty Members and Associates may unite in submitting another list to the Secretary which shall also be sent out to the membership to be considered in connection with the list already sent. In any event, the six Members who receive the highest number of votes shall be declared elected Members of Council and the two Associates receiving the highest number of votes shall be declared elected Associate Members of Council, in each case for a term of three years.

6. A vacancy in the office of President shall be filled by ballot by the Council from the list of Past Presidents, Honorary Vice-Presidents and Vice-Presidents until the end of the year in which it occurs. At the annual meeting of that year a new President shall be elected for three years in the manner prescribed in paragraph 3. A vacancy in the office of member of Council shall be filled by the Council for the unexpired portion of term of the Member or Associate causing the vacancy.

7. The President, Past Presidents, Honorary Vice-Presidents and Vice-Presidents shall be *ex-officio* members of Council.

8. The Council may hold meetings subject to the call of the President, as often as the interests of the Society may demand.

9. At all meetings of Council seven members shall constitute a quorum.

10. The Secretary and Treasurer shall be elected annually by the Council, but may be removed at any time by a majority vote of the Council after due notice has been given.

11. The Secretary must be a Member of the Society.

Article V.

Management.

1. (1) The President shall have general supervision over the affairs of the Society, appoint special committees, and preside at the annual general meetings. He shall be *ex-officio* member of all committees.

(2) In the absence of the President, one of the Past Presidents, Honorary Vice-Presidents or Vice-Presidents, in the order of seniority as determined by original accession to, or election in, their respective grades, shall preside and perform all the duties of the President; where there is the same date of seniority, the alphabetical order will govern. Provided, however, that for the Annual Meeting of the Society or for any other special occasion when it is known that the President cannot attend and preside, the Executive Committee shall, in its discretion, select and designate as Acting President any one of

the Past Presidents, Honorary Vice-Presidents, Vice-Presidents or members of Council, who shall act for the President in his temporary absence and shall perform all the duties which would devolve upon that officer during such Annual Meeting or on such special occasion.

2. (1) The direct management of the Society shall be vested in an Executive Committee of seven, composed of five members of Council, elected annually by the Council, and the President and the Secretary of the Society *ex-officio*. At least three of the five elective members of the committee shall be Members of the Society.

(2) Meetings of the Executive Committee may be held at any time, subject to the call of the Chairman; and four members shall constitute a quorum for the transaction of any business that may be properly brought before the committee.

3. The Executive Committee shall manage the affairs of the Society in conformity with the laws under which it is incorporated and the provisions of this Constitution. It shall direct the investment and care of the funds of the Society; make appropriations for specific purposes; arrange for the reading and publication of professsional papers; take measures to advance the interests of the Society, and generally direct its affairs under such regulations as the Council may from time to time prescribe.

4. The Executive Committee shall make an annual report to the Society, transmitting the report of the Secretary and Treasurer, and of any special committee which may have been ordered.

5. (1) The Secretary shall be the Executive Officer of the Society under the immediate direction of the President and the Executive Committee.

(2) He shall prepare the business for the annual meetings and record the proceedings thereof.

(3) He shall be responsible for all expenditures and certify the accuracy of all bills or vouchers upon which money has been paid, and he shall conduct the correspondence of the Society and keep full records of the same.

6. The Treasurer shall see that all money due the Society is collected and carefully invested in such manner as the Executive Committee may direct. If considered advisable by the Council, the duties of Treasurer may be performed by the Secretary.

7. The accounts of the Secretary and Treasurer shall be audited annually in such manner as the Executive Committee may direct.

Article VI.

Meetings.

1. There shall be at least one annual general meeting of the Society for the reading and discussion of professional papers, election of officers for the ensuing year, and transaction of such other business as may be brought before it. The time and location of this meeting shall be determined by the Council at least three months prior to the date fixed.

2. Special meetings may be called by the Executive Committee at the request of twenty members, which request shall state the purpose of the meeting. The call for such meetings shall be issued ten days in advance, and shall state the purpose thereof. At these meetings thirty members shall constitute a quorum.

3. The Society may adopt, from time to time, such rules as it may think proper for the order of business at its meetings.

Article VII.

Regulations and By-Laws.

1. The Council shall have authority to establish such by-laws and regulations as may be necessary for the government of the Society in the conduct of its affairs, provided that such by-laws and regulations do not conflict with the provisions of this Constitution and that they are approved by a two-thirds vote of the members of Council present at any meeting regularly called for the consideration of same. If, however, objection be made by any member of Council to a by-law or regulation so proposed, it must be submitted in writing for the action of the entire Council and will not be finally adopted unless a majority of the Council signifies its approval.

Article VIII.

Amendments.

1. Proposed amendments to the Constitution must be reduced to writing and signed by not less than ten members. They shall be forwarded to the Secretary at least ten days before the annual general meeting, and shall be immediately forwarded to the Council for its consideration. If a majority of the Council approve the proposed amendment it shall be presented to the Society at the next ensuing general meeting for discussion; if approved by two-thirds of the members present, voting by ballot, if a ballot be demanded, it shall be adopted.

INTRODUCTORY PROCEEDINGS

MINUTES OF THE TWENTY-EIGHTH ANNUAL MEETING OF THE SOCIETY OF NAVAL ARCHITECTS AND MARINE ENGINEERS, HELD AT THE ENGINEERING SOCIETIES BUILDING, NEW YORK CITY, THURSDAY AND FRIDAY, NOVEMBER 11 AND 12, 1920.

FIRST SESSION.

THURSDAY MORNING, NOVEMBER 11, 1920.

In the absence of the president, Admiral Capps, Vice-President H. L. Ferguson acting as president, called the meeting to order at 10.40 a. m.

ACTING PRESIDENT:—The meeting will please come to order. The Secretary-Treasurer will present the annual report.

REPORT OF SECRETARY-TREASURER.

NOVEMBER 8, 1920.

To the Council of The Society of Naval Architects and Marine Engineers:—

GENTLEMEN:—I have the honor to submit the following report showing the condition of the Society at the close of the fiscal year ended October 31, 1920.

The membership of the Society as at October 31, 1920, was as follows:—

Class of Members	Membership, October 31, 1919	Elected at 1919 Meeting	Promotion, Associate to Member	Promotion, Junior to Member	Promotion, Junior to Associate	Membership at Close Annual Meeting, 1920	Promotion, Members to Life Members	Promotion, Associate to Life Associates	Reinstated, 1919-1920	Deaths, 1919-1920	Resignations, 1919-1920	Suspended, 1919-1920	Membership, October 31, 1920
Members	991	128	+7	+1	—	1127	−2	—	1	8	9	14	1095
Associates	317	58	−7	—	+3	371	—	−1	—	1	14	19	336
Juniors	60	11	—	−1	−3	67	—	—	—	—	1	3	63
Life Members	16	—	—	—	—	16	+2	—	—	1	—	—	17
Life Associates	7	—	—	—	—	7	—	−1	—	—	—	—	8
Honorary Members	1	—	—	—	—	1	—	—	—	—	—	—	1
Honorary Associates	1	—	—	—	—	1	—	—	—	—	—	—	1
Totals	1393	197	—	—	—	1590	—	—	1	10	24	36	1521

FINANCIAL

RECEIPTS.

From November 1, 1919, to October 31, 1920, inclusive.

Entrance fees of new members for the year ending October 31, 1920		$1,900.00
Dues for the year ended December 31:—		
1920	$13,045.00	
1919	840.00	
1918	125.00	
1917	75.00	
1916	20.00	
1915	10.00	
		14,115.00
Dues paid in advance:—		
1921	$125.00	
1922	10.00	
1923	10.00	
1924	10.00	
		155.00
Entrance fees and dues paid in advance by proposed		85.00
Dues in arrears—reinstated members		50.00
Life membership entrance fee		1,000.00
Total receipts of entrance fees and dues		$17,305.00
Sales of publications		2,039.85
Banquet, annual meeting, 1919		3,899.05
Banquet, annual meeting, 1920, reservations		2,010.00
New York meeting, 1920, Entertainment Committee		1,600.00
New York meeting, 1919, Entertainment Committee		1,884.20
Miscellaneous receipts—sale of old papers and linoleum		139.20
Interest received:—		
On bank balances	$85.84	
On investments—other than endowment fund	120.00	
		205.84
On endowment fund investments:—		
Endowment fund additions	$272.96	
Transferred to general fund	1,122.53	
		1,395.49
Total receipts		$30,478.63
Balance—Cash in bank and on hand November 1, 1919		3,473.82
Endowment fund cash on deposit with The Bank of America, Trustee		346.09
		$34,298.54

STATEMENT.

DISBURSEMENTS.

From November 1, 1919, to October 31, 1920, inclusive.

Printing:—		
Preliminary papers, 1919		$3,609.13
Vol. 27, 1919 Transactions		8,509.68
1920 Year Book		564.00
Pamphlets		95.84
Storage of volumes		275.25
Expressage on volumes and papers		149.25
Expenses of 27th annual meeting, 1919		1,137.49
Expenses of 27th annual banquet, 1919		4,287.00
Expenses of New York meeting, 1919, Entertainment Committee		1,884.25
Rent of society rooms		840.00
Salaries of officers and clerk		2,731.61
Office expenses, postage, stationery		1,653.67
Audit of books and accounts, year ending October 31, 1919		378.75
Exchange on remittances of members		10.83
Trustee's commission—The Bank of America		46.48
Donations:—		
Ericsson Memorial	$50.00	
America's gift to France	100.00	
		150.00
Office furniture and fixtures		118.00
Total disbursements		$26,441.26
Balance, October 31, 1920:—		
On deposit with The Bank of America	$5,558.21	
Checks undeposited	1,609.00	
Drafts deposited for collection	20.00	
Cash on hand	51.02	
		7,238.23
Endowment fund cash on deposit with The Bank of America, Trustee		619.05
		$34,298.54

DEATHS (10).

Life Member (1).

H. C. Frick.

Members (8).

William Boyd.
Charles B. Calder.
J. Irving Chaffee.
Frank Jeffrey.
Edmund Mills.
William T. Nevins.
Gustav Prochazka.
Alvin A. Winship.

Associate (1).

Henry L. Gantt.

RESIGNATIONS (24).

Members (9).

Jean M. Allen.
James F. Barber.
Clarence A. Carr.
James H. Chalker.
Arthur F. Johnson.
Frank Johnson.
M. Lovatelli.
Louis P. Maier.
John K. Turnbull.

Associates (14).

William H. Becker.
James B. Bonner.
Arthur S. Goodwin.
Otto L. Halenbeck.
Morton A. Howard.
S. Victor McLeod.
Thomas I. Miller.
John Mitchell.
Percival R. Moses.
Harold L. Paige.
Donald D. Smith.
Louis M. Stoddard.
Cecil L. Thompson.
John N. Trainer, Jr.

Junior (1).

George Laing, Jr.

Statement of Resources and Liabilities as at October 31, 1920.

RESOURCES.

Cash in bank and on hand		$7,238.23
Membership dues (after deducting unpaid collectible accounts) per Schedule "1"		2,327.00
Due from dealers and members for publications		973.44
Publications on hand:—		
Volumes Nos. 1 to 18, inclusive, at 50 per cent of cost Volumes Nos. 19 to 27, inclusive, at cost		7,249.24
Office furniture and equipment		914.50
Investments:—		
$3,000—Central Pacific Ry. 1st Ref. 4% bonds at cost		2,651.25
Endowment Fund:—		
Investments at cost:		
$4,000—Baltimore & Ohio R. R. Com. 4½ bonds	$3,925.00	
$2,000—Morris & Essex R. R. 1st Ref. 3½% bonds	1,732.50	
Investments at par:		
New York City 4½% bonds due May 1, 1957	7,000.00	
Brooklyn Public Market 3½% bonds	5,000.00	
U. S. Liberty and Victory bonds	15,050.00	
	$32,707.50	
Cash on deposit with custodian	619.05	
		33,326.55
Total resources		$54,680.21

LIABILITIES.

Annual banquet, 1920, reservations		$2,010.00
Entrance fees and dues paid in advance		260.00
New York meeting, advances		1,600.00
Total liabilities		$3,870.00
Society's net worth at October 31, 1920:—		
Endowment fund, per Schedule "2"	$36,379.05	
Net worth other than endowment fund, per Schedule "3"	14,431.16	
		50,810.21
		$54,680.21

SCHEDULE "1."

Dues Outstanding as at October 31, 1920.

Summary	*Total*	*1920*	*1919*	*1918*	*1917*	*1916*	*Vol. 26*
Members	$1,627.00	$1,032.00	$340.00	$160.00	$60.00	$30.00	$5.00
Associates	615.00	430.00	140.00	40.00	5.00	.....	.....
Juniors	90.00	50.00	30.00	5.00	5.00	.....	.....
	$2,332.00	$1,512.00	$510.00	$205.00	$70.00	$30.00	$5.00

Schedule "2."

Endowment Fund as at October 31, 1920.

Particulars		*Amount*
Balance, November 1, 1919		$33,161.09
Interest earned on:—		
Municipal bonds	$490.00	
Railroad bonds	250.00	
Liberty and Victory bonds	655.49	
	$1,395.49	
Credited to general fund	1,122.53	
Balance to endowment fund	$272.96	272.96
Membership fees collected:—		
Entrance fees of new members	$1,945.00	
Permanent membership paid in cash	1,000.00	2,945.00
Total, October 31, 1920		$36,379.00

Schedule "3."

Net Worth of Society other than Endowment Fund for the year Ended October 31, 1920.

Particulars			*Amount*
Balance, November 1, 1919			$16,486.51
Additions:—			
Dues for year 1920		$14,960.00	
Dues for members reinstated		82.00	
		$15,042.00	
Less dues written off:			
Deaths and resignations	$220.00		
Failed to qualify	395.00		
Suspended	320.00		
		935.00	
			14,107.00
Sales of publications			2,104.95
Surplus on annual banquet, 1919			803.00
Miscellaneous receipts			159.20
Interest earned:			
On investments (other than endowment fund)		$120.00	
On endowment fund investment		1,122.53	
On bank balances		85.84	
			1,328.37
			$34,989.03

Brought forward			$34,989.03
Deductions:—			
Membership entrance fees for year		$1,945.00	
Less: Suspended—1920 and back years	$395.00		
Collected fees transferred to endowment fund, per Schedule "2"	1,945.00		
		2,340.00	
Total suspended fees written off		$395.00	
Cost of publications distributed to members and sold to dealers		12,174.01	
Expenses, 27th annual meeting		1,516.24	
Rent of society rooms		840.00	
Salaries of officers and clerk		2,731.61	
Office expenses, postage, stationery, etc		1,628.67	
Exchange on remittances of members		10.83	
Miscellaneous printing—membership list		564.00	
Transportation of volumes		112.28	
Storage of volumes		275.25	
Trustees' commissions		46.48	
Depreciation on office furniture—10% off $885.00		88.50	
Donations		175.00	
			20,557.87
Balance, October 31, 1920			$14,431.16

The financial statements and books of accounts have been audited by Mills & Company, Public Accountants, 42 Broadway, New York, N. Y.

Respectfully submitted,

DANIEL H. COX,
Secretary-Treasurer.

SECRETARY-TREASURER COX:—Mr. Chairman and gentlemen, the report, as usual, is distributed throughout the room. On the first page the membership of the Society is set forth. You will notice that last year we added to our membership by approximately 200. From the applications received this year, it looks as if we will get a larger increase at the present meeting than we secured last year.

With respect to the financial statement, which is shown on the following pages, as well as other information, the salient points are shown in the balance between receipts and disbursements, and also the statement of net worth. In this connection there is one matter which I desire to call attention to, and that is with the present rate of printing, office expenses and expenses of all kinds, the Society cannot continue, as it always has done, to publish its proceedings in a high-grade manner, good illustrations on good paper, and give the service that it always has given, without running behind in the operating expenses, unless there is an increase in the dues.

Later in the day the matter of a slight increase in the dues will be taken up, but this report shows very conclusively that unless there is such an increase, we cannot continue to give the service we have given. The report which is before you now is, of course, for the expenditures of the preceding year, and in that year the volume of proceedings was quite large, as there were fourteen or fifteen papers presented at the annual meeting. This year, knowing that the Society was running behind during the past year on account of the increased cost of printing, the Papers Committee very wisely reduced the number of papers to be presented at this meeting to ten, and therefore the expense of printing these papers will not be so great as in previous years, thus enabling us to make a certain retrenchment. I am sure, however, that all the members will agree that it is essential that the Society should not be placed in the position of having to refuse a good paper because some of the members may object to paying five dollars additional dues.

The endowment fund has now reached appreciable proportions. Last year it was $33,161.09. This year, by the increment of the initiation fees and the addition of one life membership, that of Mr. Andrew Fletcher, the fund stands at $36,379.00.

This endowment fund is to be automatically increased every year by all entrance fees, life membership fees, contributions of all kinds, and by the interest on its own investments, so that the fund is steadily increasing. The purpose of it is to place the Society in a position where it may, as the other societies do, from time to time do more than simply hold meetings and have papers printed, but that it may be able to give prizes for valuable papers and stimulate interest in the Society and, in a general way, participate in the undertakings which are common to the other engineering societies, which we are estopped from doing at the present time by reason of our very meager financial budget.

During the early part of the year it became evident to the Council and Executive Committee, on account of the large increase in our membership, that certain changes would have to be made in the method of conducting the affairs of the Society. You must realize that the year before last our membership increased enormously, and last year there was also a large increase. This year we also expect a large addition, so that our membership at the present time is twice what it was a few years ago.

In the old days the routine work of the Society, the writing of the necessary form letters and carrying on such work as had to be carried on was done at odd times in the evening by Mr. Kain; but it was found, on account of the large increase in membership, that all of these duties had multiplied enormously and that it was impossible to carry them on properly without having someone constantly in charge. It was further evident

that it would be to the benefit of the members, particularly to the out-of-town members, always to have the rooms of the Society open for the giving of information and any other purpose when they came to the city and visited the headquarters of the Society; so the Council and Executive Committee during the year arranged, as you all know, I think, for the office of the Society to be open at all time during business hours, in charge of Mr. Kain.

One important effect of having someone constantly on the job—the secretary endeavors to be as active as he can, but he cannot give his entire time to this work—is the improvement in the collection of the dues. There has been a marked decrease in the number of delinquent members. It will be a matter of surprise to those who pay their dues promptly to learn how hard it is to collect the dues of certain of the members.

In endeavoring to collect what is due, we first send out a regular bill, then we send out another bill, without any particular comment, and then we commence sending routine letters, each one a little more insistent than the one which preceded. After we have sent out two letters, we forward, not a peremptory, but a registered letter, and then follow with telegrams in certain cases. You would be surprised to see how those to whom the dues do not mean anything at all neglect the matter and wonder why the Society is so insistent on collecting these dues. It would assist those in charge of the affairs of the Society very greatly if our members would make a point to remit their dues promptly on the receipt of the first bill. That would eliminate the further labor incidental to sending statements and letters urging the payment of the dues.

When you come to think that the cost of the publication of the volume of proceedings is approximately $8 a volume and the annual dues are only $10, it is evident that the Society does not have a very large working capital on which to transact its business, and if on top of that any considerable number of the members fail to pay their dues, you will see at once that it is a little difficult to carry on the affairs of the Society without a certain embarrassment. The efforts of Mr. Kain, who is assisting the Secretary-Treasurer in the office of the Society in that direction, have been very helpful.

At this time we should all try to secure additional applications for membership in the Society by those who are eligible. Now is the most opportune time to do this. There are plenty of blanks on hand, and if any of our members know of anyone they would like to propose for membership, who is eligible, application blanks can be submitted and can be acted on at any time during the meeting, prior to the last session.

ACTING PRESIDENT:—I will ask the members, in accordance with our usual custom, to please rise during the reading of the names of those members who died during the past year.

Secretary Cox read the names of the members who died during the previous year, and the members of the Society stood during the reading.

ACTING PRESIDENT:—The Secretary-Treasurer will now read the names of those members elected as Vice-Presidents, Members of the Executive Committee, the Members of Council, Associate Members of Council, Committee on Papers and Life Member.

The secretary read the following announcements of election of officers and members:—

For Honorary Vice-Presidents:—W. J. Baxter and W. F. Durand.

For Vice-Presidents, term expiring December 31, 1923:—F. L. DuBosque, H. L. Ferguson, H. A. Magoun, W. A. Dobson.

For Secretary-Treasurer:—D. H. Cox.

For Executive Committee:—Stevenson Taylor, Andrew Fletcher, W. M. McFarland, F. L. DuBosque, J. W. Powell and the president and secretary-treasurer *ex-officio.*

As a result of the ballot of the membership the following were elected Members of Council, for the term ending December 31, 1923:—H. C. Sadler, D. H. Cox, C. F. Bailey, W. H. Todd, E. H. Rigg, Wm. McEntee.

For Associate Members of Council the following were elected:—H. L. Aldrich, H. H. Raymond.

For Committee on Papers:—W. M. McFarland, F.L. DuBosque, H. L. Aldrich.

The Council also elected Mr. Andrew Fletcher as a Life Member.

ACTING PRESIDENT:—I ask the secretary to read, for your action, a list of the names of the applicants for membership.

THE SECRETARY:—The following applicants for the grade of full membership, 117 in all, have been examined by the Council and recommended for election to that grade:

Members (117).

David F. Anderson, Master Mechanic, Oscar Daniels Shipyard, Tampa, Fla.

David G. Anderson, Assistant District Manager, U. S. Shipping Board, 907 Lexington Bldg., Baltimore, Md.

George F. Anderson, Inspector, Division of Construction, U. S. Shipping Board, Philadelphia. P. O. address, 4418 Chestnut Street.

Jeremiah F. Arringdale, Superintendent Squantum Plant. P. O. address, 55 Minot Street, Dorchester, Mass.

Joseph W. Atkinson, Assistant Turbine Engineer, New York Office, The Parsons Marine Steam Turbine Co., Ltd., 2 Rector Street, New York, N. Y.

Richard W. Begley, Assistant Superintendent, Booth & Co., 17 Battery Place, New York, N. Y. 1388 Bedford Ave., Brooklyn.

E. George Bernard, Engineer, F. S. Martin, 12 Broadway, New York, N. Y. P. O. address, 65 Florence Ave., Revere, Mass.

Alexander Boyt, Consulting Engineer, F. S. Martin, 12 Broadway, New York, N. Y. P. O. address, 298 Park Ave., Newark, N. J.

Louis C. Brooks, Electrical Engineer, Bethlehem Shipbuilding Company, Bethlehem, Pa.

Alfred B. Brown, Assistant Naval Architect, Bethlehem Shipbuilding Corporation, Bethlehem, Pa.

Daniel J. Brown, Chief Draughtsman, Pusey & Jones Company, Gloucester City, N. J. P. O. address, 1019 N. 64th Street, Philadelphia, Pa.

William T. Brown, Lieutenant Commander, C. C., U. S. N. R. F., Wycombe and Elder Avenues, Lansdowne, Pa.

Charles B. Buerger, Chief Engineer, Atlantic Refining Company, Philadelphia, Pa. P. O. address, 3144 Passyunk Avenue.

Edward Champness, Head Department Naval Architecture, University of Wales, "Briarbank," Dynas Powis, Glamorgan, South Wales.

Lawrence B. Chapman, Professor of Naval Architecture, Lehigh University, Bethlehem, Pa.

Johannes F. Christensen, Chief Draughtsman, Federal Shipbuilding Company. P. O. address, 221 Woodside Avenue, Newark, N. J.

Spendley Coates, Naval Architect and Engineer, Standard Oil Company of New York. P. O. address, Room 204, 26 Broadway, New York, N. Y.

Albert F. Cochran, Superintendent, Repairs and Estimator, Baltimore Dry Dock and Shipbuilding Company, 120 Broadway, New York, N. Y. P. O. address, 620 Riverside Drive, New York, N. Y.

Edward L. Cochrane, Lieutenant Commander, C. C., U. S. N., New Work Superintendent, Navy Yard, Philadelphia. P. O. address, 2528 South 20th Street, Philadelphia, Pa.

John Coleman, Chief Operating Engineer, Marine Department, The Babcock and Wilcox Company, 85 Liberty Street, New York, N. Y.

Isaiah C. G. Cooper, Draughtsman, Standard Shipbuilding Company, Shooters Island, Richmond Boro, N. Y. P. O. address, 230 St. Mark's Place, New Brighton, Staten Island.

William P. Craig, Resident Machinery Inspector, Emergency Fleet Corporation, U. S. Shipping Board. P. O. address, Manitowoc Shipbuilding Company, Manitowoc, Wis.

Frederick G. Crisp, Commander, C. C., U. S. N., General Work Superintendent, Hull Division, Navy Yard, New York, N. Y.

John F. Dalcher, Assistant Chief Engineer, Federal Shipbuilding Company. P. O. address, 31 Wegman Parkway, Jersey City, N. J.

Peter Dias, Superintending Engineer, Callaghan, Atkinson & Company, 11 Broadway, New York, N. Y. P. O. address, 1050 40th Street, Brooklyn, N. Y.

Francis R. Dravo, President, The Dravo Contracting Company, Pittsburgh, Pa. P. O. address, Sewickley, Pa.

Samuel P. Edmonds, Surveyor on Technical Staff, American Bureau of Shipping, 66 Beaver Street, New York, N. Y. P. O. address, 644 Mansfield Place, Brooklyn, N. Y.

Edward A. Edwards, Naval Architect and Engineer, Marine Department, Fairbanks, Morse & Company, 917 Arch Street, Philadelphia, Pa.

Daniel E. Erickson, Consulting Engineer, White Fuel Oil Engineering Corporation, 742 East 12th Street, New York, N. Y.

Nisbet C. Fleming, Chief Hull Draughtsman, Halifax Shipyards, Ltd., Halifax, N. S. P. O. address, 133 Windsor Street, Halifax, N. S.

William B. Flexer, Assistant Chief Draughtsman, Hull Construction Office, Wm. Cramp & Sons S. & E. B. Company. P. O. address, 4358 Manayunk Avenue, Philadelphia, Pa.

Lawrence B. Fountain, Senior Engineer Assistant, U. S. S. B., Emergency Fleet Corporation, San Francisco, Cal. P. O. address, 434 Leavenworth Street, San Francisco, Cal.

Ernest F. Frey, Hull Inspector and Surveyor, Sinclair Navigation Company. P. O. address, 385 Conarroe Street, Roxborough, Philadelphia, Pa.

Walter H. Freygang, Engineer, Walter Kidde & Co., Inc., 140 Cedar Street, New York, N. Y. P. O. address, 61 Highwood Terrace, Weehawken, N. J.

Ross L. Fryer, Consulting Engineer, Commercial National Bank Building, Washington, D. C.

William F. Gibbs, Chief of Construction, International Mercantile Marine Company. P. O. address, 9 Broadway, New York, N. Y.

Louis J. Gorar, Marine Superintendent, Iralia-America Society of Maritime Trade, Inc., 1 State Street, New York, N. Y. P. O. address, 1019 85th Street, Brooklyn, N. Y.

Kanezo Goto, Engineer Captain, Imperial Japanese Navy, Chief Inspector of the Japanese Naval Inspector's Office in America. P. O. address, 1 Madison Avenue, New York, N. Y.

John H. Gullak, Cost Engineer, Merchant Shipbuilding Corporation, Chester, Pa. P. O. address, 443 Bellevue Avenue, Trenton, N. J.

Hans Gunderud, Consulting Engineer, Ship and Engineer Surveyor, Northern Underwriting Agency, 56 Beaver Street, New York, N. Y. P. O. address, 159 76th Street, Brooklyn, N. Y.

Edgar L. Hallowell, Estimator, Pusey & Jones Company, Gloucester City, N. J. P. O. address, 302 Winona Avenue, Philadelphia, Pa.

William S. Haney, Sales Engineer, Pennsylvania Seaboard Steel Corporation, Philadelphia, Pa. P. O. address, 604 W. 9th Street, Chester, Pa.

Isaac Harter, Assistant to the President, The Babcock & Wilcox Co., 85 Liberty Street, New York, N. Y.

George P. Haynes, Chief Draughtsman, White Fuel Oil Engineering Co.'s plant of Todd Shipyards Corporation, 742 East 12th Street, New York, N. Y. P. O. address, 1467 Pacific Street, Brooklyn, N. Y.

Ernst A. Heden, Technical Director and Manager, Aktiebotaget Götaverken, Göteberg. P. O. address, Göteburg, Sweden.

John E. Hoover, Shipping Board Inspector, U. S. S. B., Flat Iron Bldg., Norfolk, Va. P. O. address, care U. S. S. B., Norfolk, Va.

George Hutchinson, President and General Manager, Newburgh Shipyards, Inc., Newburgh, N. Y. P. O. address, Palatine Hotel, Newburgh, N. Y.

William Jamin, Jr., Assistant Superintendent, Tietjen & Lang Dry Dock Co. P. O. address, 6 Grauert Place, Weehawken, N. J.

Horace C. Jefferson, General Superintendent, Green Star Line and Cosmopolitan Shipping Company. P. O. address, 475 82nd Street, Brooklyn, N. Y.

George H. Johnston, Superintendent of Outfitting, New York Shipbuilding Company. P. O. address, 432 Linden Street, Camden, N. J.

William W. Kelly, Chief Draughtsman, Tietjen & Lang Plant of Todd Shipyard Corporation, Hoboken. P. O. address, 311 Livingston Avenue, Lyndhurst, N. J.

J. H. King, Engineer, Marine Department, The Babcock & Wilcox Company, 85 Liberty Street, New York, N. Y.

Carl E. Klitgaard, Manager, Repair Dept., U. S. S. B., 45 Broadway, New York, N. Y. P. O. address, Apt. 1-A, 117 Featherbed Lane, New York, N. Y.

Norman W. Lanier, Hull Superintendent, Lake Torpedo Boat Company, Bridgeport, Conn. P. O. address, 186 Livingston Place, Bridgeport, Conn.

Edward H. LeTourneau, Assistant Manager, Repair Department, U. S. Shipping Board, 45 Broadway, New York, N. Y.

Charles E. Lucke, Professor Mechanical Engineering and Head of Department at Columbia University, New York City; Consulting Engineer, Worthington Pump and

Machinery Corporation, 115 Broadway, New York, N. Y. P. O. address, 544 West 142nd St., New York, N. Y.

Kiichiro Matsunaga, Engineer, Technical Department of Head Office of Mitsubishi Zosen Kaisha, Ltd., Tokyo, Japan. P. O. address, Mitsubishi Zosen Kaisha, Ltd., Marunouchi, Tokyo, Japan.

James D. MacBride, President, Macbee Company, 298 Broadway, New York, N. Y.

Alexander MacWatt, Surveyor, Lloyd's Registry of Shipping, 17 Battery Place, New York, N. Y. P. O. address, 31 22nd Street, Elmshurst, N. Y.

John McArdle, Manager's Representative, Bethlehem Shipbuilding Corporation, Ltd., Fore River Plant, Quincy, Mass. P. O. address, 127 Quincy Avenue, East Braintree, Mass.

William J. McAuslin, Superintending Engineer of M. H. Tracy & Co., Inc., 17 State Street, New York, N. Y. P. O. address, 355 St. John's Place, Brooklyn, N. Y.

George R. McClelland, Assistant Hull Superintendent, South Yard, New York Shipbuilding Corporation, Camden, N. J. P. O. address, 905 Oriental Avenue, Collingswood, N. J.

Ray McCoy, Chief Draughtsman, Tebo Yacht Basin Plant, Todd Shipyards Corporation. P. O. address, 136 88th Street, Brooklyn, N. Y.

Alexander McGregor, Chief Inspector, Repair Department, U. S. Shipping Board, 45 Broad Street, Boston, Mass.

James F. McLaughlin, Superintendent of Hull Construction, Newburgh Shipyard, Inc. P. O. address, 39 Townsend Avenue, Newburgh, N. Y.

Guy H. Moon, President, Moon Engineering Company, 535 Front Street, Norfolk, Va.

William P. Morehouse, General Foreman, Hoboken Marine Shops, Pennsylvania System, Hoboken, N. J. P. O. address, 94 Hauxhurst Avenue, Weehawken, N. J.

E. P. Morse, Head of Morse Dry Dock and Repair Company, Brooklyn, N. Y. P. O. address, 47 Plaza Street, Brooklyn, N. Y.

Silas Livingstone Morse, Jr., Manager, Boat Repairing Corporation, Pier 11, North River, New York, N. Y. P. O. address, 119 West 37th Street, Bayonne, N. J.

William Mulheron, Assistant Chief Engineer, Merchant Shipbuilding Corporation, Chester, Pa. P. O. address, 137 Rutgers Avenue, Swarthmore, Pa.

Joseph J. Nelis, Secretary, Foster Marine Boiler Corporation, 111 Broadway, New York, N. Y.

William Nexsen, Boiler Engineer, Federal Shipbuilding Company, Kearny, N. J. P. O. address, 38 Madison Avenue, Jersey City, N. J.

Harry Nightingale, Superintendent of Works, Todd Shipyards Corp. (Plant of), White Fuel Oil Engineering Corp., 742 East 12th Street, New York, N. Y. P. O. address, 311 DeMott Street, West Hoboken, N. J.

Hugo F. Nordström, Docent of the Royal Technical College, Stockholm. P. O. address, Datagatan 39, Stockholm, Sweden.

Robert E. Payne, Chief Engineering Draughtsman, Geo. G. Sharp, N. A., New York, N. Y. P. O. address, 121 Garrison Avenue, Jersey City, N. J.

John Perkins, Ship and Engine Surveyor, American Bureau Shipping, 66 Beaver Street, New York, N. Y. P. O. address, 96 Stuyvesant Avenue, Arlington, N. J.

Charles T. Perry, Mechanical Engineer, Department of Plant and Structures, City of New York. P. O. address, 224 Wardwell Avenue, Westerleigh, Staten Island, N. Y.

Oscar J. Peterson, Chief of Engineering, Scientific Department, New York Shipbuilding Corporation, Camden, N. J.

Harry P. Phelps, Assistant General Manager, Bethlehem Shipbuilding Corporation, Sparrows Point, Md.

Hannon M. Power, Marine Superintendent, Baltimore Steamship Company, Baltimore, Md.

Melville W. Powers, Lieutenant, C. C., U. S. N., Assistant to Superintending Constructor, U. S. N., N. Y. Shipbuilding Corporation, Camden, N. J.

Joseph F. Rawlinson, Superintendent, Toledo Shipbuilding Company, Toledo, Ohio. P. O. address, 658 Georgia Avenue, Toledo, Ohio.

A. C. Rimmer, Consulting Engineer and Naval Architect, 149 Broadway, New York, N. Y.

John W. Ritchie, Chief Engineer, New England Fuel & Transportation Co., Boston, Mass. P. O. address, 59 Pearl Avenue, Beachmont, Revere, Mass.

Anders M. Rodberg, Inspector of Machinery, U. S. Shipping Board. P. O. address, P. O. Box 1772, Tampa, Fla.

James Ross, Surveyor, American Bureau of Shipping, 635 Bourse Building, Philadelphia, Pa. P. O. address, 3119 South Constitution Road, Fairview, N. J.

Charles F. Scott, Commercial Engineer on Marine Work for the General Electric Company, 120 Broadway, New York, N. Y.

Edmund S. Sickles, Surveyor, U. S. Bureau of Survey, 23 Liberty Street. P. O. address, 419 64th Street, New York, N. Y.

Robert P. Sherman, Naval Architect for the Standard Oil Company of New York, North China Territory. P. O. address, care Standard Oil Co. of New York, Shanghai, China.

John T. Spencer, Charge Draughtsman, Hull Division, Standard Shipbuilding Corporation, Shooters Island, N. Y. P. O. address, 174 Perry Avenue, West New Brighton, S. I., N. Y.

Harry A. Stendrup, Assistant Marine Superintendent, Sinclair Navigation Company, 120 Broadway, New York, N. Y. P. O. address, 58 Union Street, Bath, Me.

Robert B. Stirling, Shop Superintendent, White Fuel Oil Engineering Corporation, 742 East 12th Street, New York, N. Y. P. O. address, 43 West 46th Street, Bayonne, N. J.

John C. Sweeny, Chief Engineer, S. S. Ablanset, Norton, Lilly Company, 26 Beaver Street, New York, N. Y. P. O. address, Hartsdale, N. Y.

Harry B. Taylor, Chief Turbine Engineer, Repair Section, U. S. Shipping Board, 45 Broadway, New York, N. Y. P. O. address, "Graylock," 50 Orange Street, Brooklyn.

Francis J. Tucker, General Manager, Merchant Shipbuilding Corporation, Chester, Pa.

William T. Turner, New York Agent for Crichton Thompson & Co., Ltd., 17-19 Cockspur Street, London, S. W. P. O. address, 44 Whitehall Street, New York, N. Y.

Ingebrigt Gunerius Volden, Chargeman (Hull and Rigging), Newburgh Shipyards, Newburgh, N. Y. P. O. address, 28 Liberty Street, Newburgh, N. Y.

Cullen H. Want, Assistant General Manager, Morse Dry Dock & Repair Company, Ft. 56th Street, Brooklyn, N. Y.

Walter W. Watson, Assistant Engineer (Naval Work), Bethlehem Shipbuilding Corporation, Bethlehem, Pa. P. O. address, 146 East Broad Street, Bethlehem, Pa.

James R. White, Assistant Naval Architect, Bethlehem Shipbuilding Corporation. P. O. address, Bethlehem Club, Bethlehem, Pa.

Norman C. Wiley, Assistant Chief Merchant Hull Draughtsman, Bethlehem Shipbuilding Corporation, Bethlehem, Pa. P. O. address, P. O. Box 64, Bethlehem, Pa.

Daniel Wilkie, General Superintendent, Ore Steamship Corporation, 111 Broadway, New York, N. Y. P. O. address, 190 Major Avenue, Arrochar, Staten Island, N. Y.

William S. Wilkinson, Assistant Engineer, Bethlehem Shipbuilding Corporation, Bethlehem, Pa. P. O. address, 33 Club Avenue, Bethlehem, Pa.

Jonathan A. Wilson, General Superintendent, Outfitting, A. I. S. C., Hog Island, Pa. P. O. address, 917 13th Avenue, Moore, Pa.

Frank B. Worden, Assistant to Manager of Construction in Charge of Great Lakes District, Emergency Fleet Corporation, 140 North Broad Street, Philadelphia, Pa.

Walter W. Young, Resident Engineer, Row & Davis Engineers, Inc., 90 West Street, New York, N. Y. P. O. address, 902 Liberty Building, Philadelphia, Pa.

ACTING PRESIDENT:—Gentlemen, you have heard the list of proposed new members of the Society which has been read by the secretary. What is your pleasure?

PROFESSOR SADLER:—I move that the gentlemen whose names have just been read be duly elected to membership in the Society.

The motion was duly seconded, put to vote, and unanimously carried.

ACTING PRESIDENT:—The secretary will now read the list of applicants for election as Associate Members of the Society.

The secretary read the list of applicants for election to Associate Membership.

Associate Members.

Edmund D. Bistline, Works Accountant, Federal Shipbuilding Company, P. O. Box 618, Newark, N. J. P. O. address, 338 Roseville Avenue, Newark, N. J.

Stephen C. Cafiero, Chief Draughtsman, Standard Shipbuilding Corporation. P. O. address, 38 Tompkins Place, Brooklyn, N. Y.

Harry H. Chenoweth, Assistant Superintendent, Floating Equipment, Penn. Railroad Company. P. O. address, Pennsylvania Terminal, New York City (Room 718).

Hans A. Christensen, Draughtsman, Federal Shipbuilding Company. P. O. address, 71 Millington Avenue, Newark, N. J.

Maurice J. Cotter, Draughtsman, International Mercantile Marine Company, 1 Broadway, New York, N. Y. P. O. address, 130 W. 63rd Street, New York, N. Y.

William J. Driver, Consulting Marine Engineer, The American Standard Shipfittings Corporation, 115 Broadway, New York, N. Y. P. O. address, 547 West 123rd Street, New York, N. Y.

Albert E. Eldridge, Treasurer, George Lawley & Son's Corporation, Neponsit, Mass. P. O. address, Neponsit, Mass.

David D. Faris, Assistant Manager, Marine Department, Westinghouse Electric & Manufacturing Company, East Pittsburgh, Pa.

Frank E. Fellows, Inspector, U. S. Shipping Board, Washington, D. C. P. O. address, 802 Chestnut Street, Massillon, Ohio.

Paul S. Grierson, Chief Engineer, Chas. Cory & Son, Inc. P. O. address, 16 Lindsley Avenue, South Orange, N. J.

Tony L. Hannah, Lieutenant (T), C. C., U. S. N., Wm. Cramp's Sons S. & E. B. Co., Philadelphia, Pa.

Walter R. Harper, Resident Representative, U. S. Shipping Board, Division of Construction, in Charge Concrete Shipyard, Mobile, Ala. P. O. address, U. S. Shipping Board, Division of Construction, 816 Howard Avenue, New Orleans, La.

Jacob F. Heigis, Assistant Superintendent, Tietjen & Lang Dry Dock Co. P. O. address, 54 11th Street, Hoboken, N. J.

Carlos Hevia, Director General of Co. de Fomento Maritimo, Empedrado 22, Havana, Cuba.

Warren E. Hoffman, Assistant, Marine Department, General Electric Company, Schenectady, N. Y.

Charles Holcomb, Vice-President, Chas. D. Durkee & Co., 2 South Street, New York, N. Y.

Alfred W. Hough, Estimator, Designing Department, Electric Boat Company. P. O. address, 19 Meridian Street, New London, Conn.

Chester D. Hubbard, Superintendent, Service Department, Bethlehem Shipbuilding Corporation. P. O. address, 715 Avenue H, Bethlehem, Pa.

William F. Hutaf, Estimator, Tietjen & Lang Steamship Co., 17th Street and Park Avenue, Hoboken, N. J. P. O. address, 56 11th Street, Hoboken, N. J.

Thomas A. Johnson, Secretary-Treasurer and Manager, The Bruce Dry Dock Company. P. O. Box 1520, Pensacola, Fla.

Joseph S. Jones, Marine Electrical Engineer and Assistant to Vice-President, Chas. Cory & Son, Inc., 183 Varick Street, New York, N. Y.

Lail T. Kane, Chief Engineer, Motorship Mazatlan, California & Mexico S. S. Company. P. O. address, Piedmont Apartments, San Pedro, Cal.

Arthur S. Kemman, Inspector, Union Oil Company. P. O. address, 1639 West 45th Street, Los Angeles, Cal.

Otto V. Knierim, Inspector of Machinery with U. S. Shipping Board at Moore Shipbuilding Company, Oakland, Cal. P. O. address, 3025 Grove Street, Berkeley, Cal.

John E. Lebret, Assistant to Naval Architect, American International Shipbuilding Corporation, Hog Island, Pa. P. O. address, 4521 N. 11th Street, Philadelphia, Pa.

Carl E. Lindquist, Draughtsman, Bethlehem Shipbuilding Corporation. P. O. address, 582 Broad Street, East Weymouth, Mass.

Albert C. Ludlum, New York Engineering Company, 2 Rector St., New York, N. Y.

David A. Lundquist, Draughtsman, Navy Yard, Boston, Mass. P. O. address, 36 Mt. Ida Road, Dorchester, Mass.

Gustav F. S. Mann, Engineer, Production Department, Newburgh Shipyards. P. O. address, 45 High Street, Newburgh, N. Y.

Rufino Martinez, Union Construction Company, Oakland, Cal. P. O. address, Berkeley, Cal.

Walter H. Michael, Superintendent Order Department, Bethlehem Shipbuilding Corporation. P. O. address, 517 Lincoln Avenue, Bethlehem, Pa.

Ernest L. Mills, Engineer, Klaxon Company, Industrial Division. P. O. address, Metropolitan Tower, New York, N. Y.

Alonzo B. Pouch, President, American Dock Company, 17 State Street, New York, N. Y.

William H. Reed, Electrical Engineer, Tietjen & Lang Dry Dock Company. P. O. address, 27 Fairfield Avenue, South Norwalk, Conn.

Manuel de Régules, Draughtsman, Downey Shipbuilding Company. P. O. address, 1035 Multnomah Street, Portland, Oregon.

Philip L. Rhodes, New York Representative, Union Shipbuilding Company, Baltimore, Md. P. O. address, 50 Church Street, New York, N. Y.

William G. Robinson, Assistant Superintendent, Tebo Yacht Basin Company. P. O. address, 188 82nd Street, Brooklyn, N. Y.

Frederick P. Rose, Sales Manager, New York Engineering Company, 2 Rector Street, New York, N. Y. P. O. address, 798 South 12th Street, Newark, N. J.

William A. Sandberg, Draughtsman, Southwestern Shipbuilding Company, East San Pedro, Cal. P. O. address, Arlington Hotel, Long Beach, Cal.

Thor G. Sandgren, Assistant Naval Architect, Standard Shipbuilding Corporation. P. O. address, 33 Fairview Avenue, West New Brighton.

George W. Selby, Treasurer and Managing Director, Marine Decking & Supply Company. P. O. address, 1011 Chestnut Street, Philadelphia, Pa.

Andrew Shuttleworth, Assistant Engineer, S. S. Sartartia, Kerr Steamship Co., 434 48th Street, Brooklyn, N. Y.

Norris R. Sibley, Marine Representative, Turbine Engineer, Westinghouse Electric & Manufacturing Company, 165 Broadway, New York, N. Y.

Richard Soderberg. P. O. address, 3130 Kearsarge Avenue, Camden, N. J.

George W. Spencer, Assistant Superintending Engineer, International Mercantile Marine Company. P. O. address, 565 81st Street, Brooklyn, N. Y.

H. Stanley Starling, Material Inspector to the Bureau Veritas, 17 State Street, New York, N. Y.

Erling B. Stockmann, Superintendent, Astmahco Navigation Company, 341 Madison Avenue, N. Y. P. O. address, 80 Vine Street, Bridgeport, Conn.

Harry J. Summers, Surveyor, American Bureau Shipping, 428 Union Oil Building, Los Angeles, Cal. P. O. address, 1244 Irolo Street, Los Angeles, Cal.

David B. Sutton, Manager Sales, Thacher Propeller & Foundry Corporation. P. O. address, 92 Willett Street, Albany, N. Y.

James T. Sweeney, Outside Superintendent, Robbins Dry Dock & Repair Company, Erie Basin, Brooklyn, N. Y. P. O. address, 249 13th Street, Brooklyn, N. Y.

Henry Taylor, Draughtsman, Newburgh Shipyards, Care Y. M. C. A., Newburgh, N. Y.

William A. Thomas, 3rd, Manager, Marine Governor Department, Chas. Cory & Son, Inc., 183 Varick Street, New York, N. Y.

Frederick S. Titsworth, Associated with Walter Kidde & Company, Inc. P. O. address, 140 Cedar Street, New York, N. Y.

Claude L. Turner, Inspector, U. S. Shipping Board, 45 Broadway, New York, N. Y. P. O. address, Cornell Club, 30 West 44th Street, New York, N. Y.

J. W. J. van Haersolte, Director, Nautical Technical Museum, Rotterdam, Holland. P. O. address, 68 Haringoliel.

Bowden Washington, Chief Engineer, Cutting & Washington Radio Corporation, 6 West 48th Street, New York, N. Y.

Wallis C. Watson, Assistant Marine Engineering Department, General Electric Company, Schenectady, N. Y.

Edward H. Weatherspoon, Electrical Engineer in Charge of Marine Installations, Chas. Cory & Son, Inc., 183-187 Varick Street, New York, N. Y.

Frank Weber, Lieutenant (T), C. C., U. S. N., Wm. Cramp's Sons S. & E. B. Co. Care Superintending Constructor's Office, Philadelphia, Pa.

Francis J. Whelan, Draughtsman, Newburgh Shipyards. P. O. address, Care Kyle & Purdy, Inc., City Island, N. Y.

Charles G. Wilkinson, General Manager, Georgia Shipbuilding Company. President, Wilkinson Machine Co., Savannah, Ga.

John Lyell Wilson, Surveyor, American Bureau of Shipping, 66 Beaver Street, New York, N. Y. P. O. address, 91 Kenmore Place, Brooklyn, N. Y.

Franklin Washington Wood, Vice-President and Consulting Engineer, Chas. Cory & Son, Inc., 183-187 Varick Street, New York, N. Y. P. O. address, 5 Irwin Park, Montclair, N. J.

Acting President:—Gentlemen, you have heard the list of applicants for election as Associate Members. What is your pleasure?

Mr. C. F. Bailey, *Member*:—I move that these gentlemen be elected as Associate Members.

The motion was duly seconded, put to vote, and unanimously carried.

Acting President:—We will now have the list of applicants recommended for election as Junior Members.

The secretary read the list of applicants for election as Junior Members.

Juniors (9).

Robert Freebairn, Assistant Naval Architect, Sinclair Navigation Company, 120 Broadway, New York, N. Y.

Theodore G. Grier, Draughtsman, Wm. Cramp & Sons S. & E. B. Co. P. O. address, 432 Trenton Avenue, Camden, N. J.

Joseph Haag, Jr., Tebo Yacht Basin Co., 23rd Street, Brooklyn, N. Y. P. O. address, 625 West 156th Street, New York, N. Y.

Carlos Krebs, Student Engineer, Union Shipbuilding Company. P. O. Address, 103 West Monument Street, Baltimore, Md.

Robert W. Macdonald, Inspector, Munson Steamship Line. P. O. address, 843 Boulevard East, Weehawken, N. J.

Howard A. McDonell, Engineer, Ship Stabilizer Department, Sperry Gyroscope Company. P. O. address, 36 Pierrepont Street, Brooklyn, N. Y.

William R. Nichols, Lieutenant, C. C., U. S. N., Massachusetts Institute of Technology, 46 Park Vale Avenue, Allston, Mass.

William A. Sullivan, Lieutenant, C. C., U. S. N., Assistant Construction Superintendent, Navy Yard, Portsmouth, N. H.

Marvin W. Wickham, Draughtsman, New York Shipbuilding Corporation. P. O. address, 1530 Collings Road, Camden, N. J.

Acting President:—Gentlemen, you had heard this list which has been read by the secretary. I understand the Juniors are not elected by the Society.

THE SECRETARY:—The Council has acted favorably upon the following transfers to various grades from Junior to Associate, from Associate to Member, and from Junior to Member.

Junior to Associate (4).

Haldwell S. Colby, Vice-President, United Machine & Manufacturing Company, Canton, Ohio. P. O. address, 217 18th Street, N. W., Canton, Ohio.

Irving Fineman, Lieutenant, C. C., U. S. N., Cramp's Shipyard, Philadelphia, Pa.

Robert B. Lea, Marine Department, Sperry Gyroscope Company, 40 Flatbush Avenue Extension, Brooklyn, N. Y.

Edward G. Sperry, Engineer for Sperry Gyroscope Company. P. O. address, 40 Flatbush Avenue Extension, Brooklyn, N. Y.

Associate to Member (7).

Harvey E. Barnett, Hull Scientific Draughtsman, Wm. Cramp & Sons, S. & E. B. Co., Philadelphia, Pa. P. O. address, 2518 N. Sartain Street, Philadelphia, Pa.

M. Thomas Green, President, General Manager and Assistant Treasurer, Richard T. Green Company, 211 Marginal Street, Chelsea, Mass.

Arnijot F. C. Grontoft, Cost Engineer, Federal Shipbuilding Company, Kearny, N. J. P. O. address, Box 618, Newark, N. J.

Max G. P. Hansen, Head of Firm: Max Hansen, N. A. P. O. address, 1532 Hazel Avenue, Beaumont, Tex.

Leigh R. Sanford, Resident Inspector, Manitowoc Shipbuilding & Burger Boat Company, Manitowoc, Wis.

Ralph M. Smith, Naval Architect, Hanlon Dry Dock & Ship Company, Oakland, Cal.

Albert R. Ware, Ship Draughtsman, Hull Scientific Department, New York Shipbuilding Corporation, Camden, N. J. P. O. address, 1517 Wildwood Avenue, Camden, N. J.

Junior to Member (6).

Harold T. Bent, Inspector, Newport News Shipbuilding & Dry Dock Company, Newport News, Va. P. O. address, 316 56th Street, Newport News, Va.

Ray E. Brown, Ship Draughtsman, Hull Scientific Department, New York Shipbuilding Corporation, Camden, N. J. P. O. address, 2934 Yorkship Square, Camden, N. J.

Arthur W. Gunn, Charge Draughtsman, Lake Torpedo Boat Company, Bridgeport, Conn. P. O. address, 87 East Main Street, Stratford, Conn.

Alton A. Norton, General Foreman of Shipway, Sun Shipbuilding Company, Chester, Pa. P. O. address, 934 East 16th Street, Chester, Pa.

Linn M. Rakestraw, Assistant to Assistant Manager, Repair Department, U. S. Shipping Board. P. O. address, 17 29th Street, Beechhurst, L. I.

Carl H. Young, Surveyor, American Bureau of Shipping, 66 Beaver Street, New York, N. Y. P. O. address, 2475 University Avenue, Bronx, N. Y.

ACTING PRESIDENT:—The next item in the order of business is the proposed amendment to the Constitution which has been unanimously approved by the Council increasing the annual dues. The reasons for this you are familiar with, but it is quite probable that the secretary will make you more familiar with them.

THE SECRETARY:—With respect to the increase, perhaps I have said all that is necessary when I discussed the matter in connection with the budget in the annual report of the Secretary-Treasurer.

When it became so evident that this increase was necessary, the Council canvassed the situation among the membership and finally took it up with the Executive Committee, and both were unanimously in favor of raising the annual dues by an amount of five dollars in each grade. The provisions of the Constitution required that ten days prior to the annual meeting an application shall be made to the Council signed by at least ten members, stating in direct terms what the proposed amendment shall be. This application has been made and presented, and the Council, at its meeting held yesterday, unanimously recommended its adoption by the Society.

The notice to the Council is as follows:—

"NEW YORK, *November 1, 1920.*

"*To the Council of The Society of Naval Architects and Marine Engineers.*

"In accordance with Article VIII of the Constitution of the Society, the following amendments to the Constitution are recommended for adoption:

"*Article III, Paragraph 1.*

"Eliminate the word 'ten' and substitute the word 'fifteen'; eliminate the word 'five' and substitute the word 'ten,' the paragraph to read as amended:

"'1. The entrance fees, payable on admission to the Society, shall be as follows:—

"'Members and Associates, fifteen dollars; Juniors, ten dollars; Honorary Members and Honorary Associates, no fees.'

"*Article III, Paragraph 2.*

"Eliminate the word 'ten' and substitute the word 'fifteen'; eliminate the word 'five' and substitute the word 'ten,' the paragraph to read as amended:—

"'2. The annual dues shall be as follows:

"'Members and Associates, fifteen dollars, Juniors, ten dollars; Honorary Members and Honorary Associates, no dues.

"*Article III, Paragraph 4.*

"Eliminate the words 'but annual dues for the current year shall not be required unless,' making a period after the word 'fee' and ending the sentence. Substitute the word 'if' before the word 'he,' beginning a new sentence. Strike out the words 'that year, in which case' and substitute the words 'the meeting at which he is elected.' Eliminate the word 'five' and substitute the word 'ten.' The paragraph to read as amended:—

"'4. The annual dues shall be payable in advance on the first day of January. The Secretary shall notify each member of the amount due for the ensuing year at the time of giving notice for the annual meeting. On notification of his election each member shall pay his entrance fee. If he desires to receive the published transactions of the Society for the meeting at which he is elected, he shall pay ten dollars in addition to the entrance fee.'

"*Article III, Paragraph 5.*

"Eliminate the words 'two hundred dollars' and substitute the words 'three hundred dollars,' the paragraph to read:—

"'5. Members and Associates can compound for all future dues and become Life Members or Life Associates by making a single payment of three hundred dollars and signing an agreement to conform to any future amendments to the Constitution and By-Laws.'

"The increase in entrance fees and annual dues provided for in above amendments, paragraphs 1 and 2, Article III, have already been approved by mail vote of the Council, in order to provide increased revenue to meet the increasing expenses of the Society. The amendments to paragraphs 4 and 5 are consequent upon those to paragraphs 1 and 2. The proposed modification of paragraph 4 is in order to make it clear to new members, many of whom have in the past been of the opinion that the payment made by them on election is for annual dues. Each year the secretary has to write numerous letters explaining the true interpretation of the paragraph as it now stands.

Stevenson Taylor	Daniel H. Cox
C. A. McAllister	Bruno Tornroth
David Arnott	Irving Cox
Joseph Hecking	Morris Douw Ferris
John Martin	W. M. McFarland

The Secretary:—As I have stated, this has been presented in proper form according to the Constitution and unanimously recommended by the Council for your adoption, at its meeting held yesterday.

Acting President:—Gentlemen, you have heard the proposed amendment, which has been approved by the Council. What is your pleasure?

Mr. W. M. McFarland, *Honorary Vice-President:*—I move that the amendment be adopted.

The motion was duly seconded.

Acting President:—Gentlemen, you have heard the motion, duly made and seconded. Is there any discussion? All in favor of the adoption of the amendment as read by the secretary signify by saying "Aye." Opposed "No." It is unanimously carried and it is so ordered.

The next item in the order of business scheduled is the opening address of the president.

The president of the Society, Admiral Capps, has asked me to say to you that he deeply regrets that his absence on an official visit to the west coast prevents him from being here throughout the meetings this year. I am sure you do not expect me, as acting president, to make a suitable or extended opening address today.

ACTING PRESIDENT'S ADDRESS.

During the past year there has been a gradual and sure development of comparatively new ideas in design and building. Oil burning in steam boilers has more and more tended to replace coal, and, on account of the greater convenience and lessened labor, has tended to become a necessity. You will have a paper on oil burning, for instance, at this meeting. Oil burning has been adopted for reasons of economy, in some cases

for reasons of convenience, and has become so generally installed in vessels that it is the rule rather than the exception. As with many other so-called luxuries, the men who go to sea now choose vessels that burn oil, so that it is rapidly becoming a necessity. A brief talk with any seagoing engineer or captain will demonstrate that oil burning, like the automobile, has probably come to stay, certainly as long as the oil supply lasts. Its labor-saving qualities and great convenience in handling is such that it is virtually a necessity at this time.

Another development, fairly recent so far as its success goes, is in connection with the geared turbine. For many years the geared turbines have been successfully used abroad, and in this country on government work, notably on destroyers. During the war we had a number of involved experiences with geared turbines—a good many of them did not work largely for the reason that other things do not work—we started out to make up 100 or 150 or 200 of something mechanical without first making a sample, and with the usual results. I am sure that none of the older engineers present would say that anything of a mechanical construction has ever worked the first time it was made. A great many different concerns make gears successfully, so that the success of the geared turbine at this time is assured, notwithstanding the fact that a great deal of trouble has been caused in some cases through the use of the geared turbine.

I would like to point out, though, that we are not the only people who have had difficulty with geared turbines in our ships. The most unsuccessful geared-turbine job that came into Hampton Roads during the war was one built and installed by some of the most notable builders abroad. I do not think that any geared turbine has ever come our way that had so much the matter with it as this particular one had.

A development in which we are all more interested now than probably any other is that of the internal-combustion engine. Every large shipbuilding concern in this country is getting ready to build, or is already building an internal-combustion engine, with a view to meeting an inevitable demand for economical fuel consumption. I should say that the ability of the internal-combustion-engine ship to make a very long trip without refueling is one of the most important features connected with this type of engine.

The electric drive is another development which is steadily coming forward. It has become a demonstrated fact in the case of a collier for the Navy and also a battleship, although there have been several minor setbacks. It is being adopted to a certain extent in merchant vessels as well.

In the matter of shipbuilding there have been no particularly new developments, as most of the shipbuilding plants have remained where they were a year ago, but there has been a gradual growth frequently unobserved in the use of the oxyacetylene welding and cutting processes and the electric-welding process. I think that is the most remarkable development in shipbuilding in recent years. It has not all been a clear gain. Some of us can remember the days when, if holes were not fair in place, it was necessary to get a hand reamer and gouge and fit them in place. With the invention of the pneumatic tool, which was going to save us much labor, we found the amount of unfairness of holes was increased enormously. With the increase in pneumatic machinery there has been a gradual falling off in the character of work done, so that the oxyacetylene process which cuts off corners, and the electric-welding process which welds together joints, repairs defects in castings and many other things, some of us have noticed, has also brought with it a great deal of work which we got along without under former conditions,

and for a large part we are busily engaged in repairing defects. The use of oxyacetylene in making repairs and the use of electric welding, both on the ships and in the plants, are tremendous, and it is impossible to estimate the amount of time and money which is saved in making repairs, both of ship work and plant work, by these processes.

While the outstanding lesson to our profession of the great war was the tremendous economic losses incurred by waiting for the war to build our marine, architects and engineers learned other lessons also. The great value of standardized design, not only of simple cargo vessels but of destroyers as well, was clearly demonstrated as never before.

In the destroyer program many builders thought they had found not only a greater advantage in standardization and in simpler types of vessels, but also that in the high-grade vessels we can more nearly compete with foreign shipyards than in the cheaper types. It has been generally assumed that a cheaper type of vessel may be manufactured, as it were, and that therefore American methods, so called, would give us an advantage.

The fallacy of that argument lies in the fact that ships are built and not manufactured, and every engineer who is not thoroughly imbued with that idea needs to become imbued with it, because otherwise there is no way of answering the statement of those, sometimes in the profession and frequently out of it, that we do things as cheaply as they may be done in any other country, so why not build ships as cheaply? A brief examination and study of the difference between manufacturing and building will show. Manufacturing or repetitive processes in shipyards probably do not involve more than 5 per cent of the labor.

During the war many men were trained, and with remarkable ease, to perform the simpler operations both in shipbuilding and in ship operation; and the character of work done by them was remarkably good under the conditions. A great deal of fun has been poked at the so-called fabricated vessels, but so far as the work done in the fabricated plants is concerned, it has been, in my judgment, far better than could reasonably have been expected; and although there have been many complaints about the operation of vessels and the lack of care, it would seem to me as if they had done much better than anyone would have thought possible.

The lack of men both to build and operate ships was due to the same causes that led to the tremendous financial loss in building our marine—we were not prepared before the war was on us.

At the present time probably half of the men who were engaged in shipbuilding during the war have been laid off. I do not know the exact figures, but there are large increases from week to week. There will be a further inevitable decrease in shipbuilding by private concerns, due not only to the lack of new orders, which is common to all manufacturing and building business at the present time, but also to the fact that during the war practically all of the navy yards in the United States were equipped for building large and small naval vessels.

Except on the Lakes, and for a comparatively small number of coastwise vessels, naval shipbuilding has been the backbone of the shipbuilding business in this country up to a short time before the great war. At the present time five navy yards are equipped for building capital ships and a large number for building the smaller craft. This, together with the lack of orders for marine vessels, will inevitably result in the laying off of many more men. When prices have fallen sufficiently to attract orders from owners, there will

probably be a number of new orders which are now in contemplation, chiefly from owners of coastwise fleets, but there will not be enough to keep more than half of the shipyards of the country busy.

At the present time the Shipping Board has many more vessels of certain types than can be operated successfully by them. It is believed now that the most important single thing connected with our business, the most important thing that can be done for a merchant marine, is to have the Shipping Board built up to full strength and so organized that it can really perform the functions it was originally intended the Shipping Board should perform.

The act creating the Shipping Board states that the board shall examine and study and determine in its own mind upon a standard policy to be followed by the United States in the acquisition and operation of a merchant marine. Apparently up to the present time the board has been so busy with the details of its business that it has not had time or opportunity to make this report, which is possibly the most important thing that it could do.

Legislative action taken by us may not be entirely agreeable to those who are firmly entrenched in the business, but it is quite probable that no other action looking to the same result of getting our fair share, our permanent share in the carriage of our own goods, would be very much more agreeable.

While this Society was founded to promote and foster the art of shipbuilding, it must be borne in mind that there must needs be something to practice on if the art is to remain alive with us, and I for one do not believe we will ever have a merchant marine in the United States without some protection to get it going.

A cause of congratulation—in fact, several causes of congratulation—to those who are in our profession may be mentioned:—The growth and development during the past year of the American insurance business, having to do with the insurance of vessels and cargoes has been very satisfactory, and, on the whole, remarkable. The growth and development of an American classification society which bids to become really great is also a cause for rejoicing, and the establishment of branches practically all over the world by the largest banking institutions in the United States, particularly of New York, is something that Americans interested in shipping may be well pleased with. All of these things are necessary to our commerce and are an essential and component part of it, and no marine has ever yet really existed in a permanent way without these necessary adjuncts.

On the whole, a great work has been done by starting our boys and citizens again upon the sea, and in interesting many responsible Americans, who ordinarily have taken no interest in maritime affairs, in ship operation and in shipowning. If the operation of American ships is reasonably profitable, there will be no lack of either men or money to purchase and operate these ships, and there is no reason why, under such circumstances, the ancient heritage of the seas may not be ours again. In these days of peace and talks of peace, and of leagues, it is well to remember one very essential fact—that a great partnership is involved, primarily a partnership between Great Britain and the United States, call it what you will; and if we are to be partners in deed and in truth, as well as in name, it is necessary in that partnership, as in any other partnership, that each member—each party to the compact—shall be as independent of the other party as may be, otherwise the partnership will become galling to that partner who depends unduly upon the other. Therefore it is well that we should recognize, and that our friends and blood

relations should also recognize, that the United States, to be an agreeable partner in any great world enterprise, must have those things which rightfully belong to it.

ACTING PRESIDENT:—The first paper, No. 1, is entitled "University Education in Ship Construction and Marine Transportation," by Prof. Lawrence B. Chapman, Member.

Professor Chapman presented the paper.

UNIVERSITY EDUCATION IN SHIP CONSTRUCTION AND MARINE TRANSPORTATION.

By Professor Lawrence B. Chapman, Member.

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As a contribution to America's merchant marine, a course in naval architecture has recently been inaugurated at Lehigh University along lines widely different from other university courses. The main feature which differentiates this course from others offered in this country and abroad is the introduction of a substantial amount of economics and business administration subjects. These subjects are built upon a solid basis of engineering education.

The aim of the course is to prepare men to meet the future requirements of shipbuilding and shipping—men well trained in engineering and the elements of shipping, yet who will be prepared, if the opportunity offers, to enter executive positions in these fields. If the engineering student is given a broad training in the fundamentals of engineering, applied economics and business management, then spends several years following his graduation in practical work and in engineering designing and construction, and at the same time keeps in touch with the problems of management and economics, he will not hesitate to assume the larger responsibility of executive work when opportunity offers.

In the scheme of studies the course follows generally the engineering courses at Lehigh, except that, as already noted, business administration subjects have been made prominent.

In introducing these new subjects to the curriculum we have not slighted in the least the basic engineering studies. We have, however, dropped out much of the advanced specialized work often included in courses in naval architecture. For instance, in marine engineering, the theory and practice of steam, Diesel and marine engineering is given vigorous treatment, yet little time is spent on reciprocating engine and turbine design—much of which is of an empirical nature. The method of design and those features that require theoretical treatment are covered, but the details and empirical methods are left for the student who wishes to follow marine engineering to learn in the industry, where they can be taught in a more thorough and satisfactory manner than in the university.

In such subjects as ship design, however, we carry through a design of a cargo ship in order to drive home the principles of naval architecture and ship construction and also to make sure that the student is thoroughly familiar with the principal tool of any branch of shipping he may enter—the ship.

Considerable time is saved in subjects of this nature by dividing up the calculations among several students. For instance, in obtaining the stability curves

for a ship, each student works out the complete calculation at several displacements for one angle of inclination. Each student is assigned a different angle and thus, with the work divided between a group of students the complete calculation is obtained. The work of all the students is then assembled and given to the class to draw up the cross curves and curve of statical stability. As the body plans used for one group have all been traced from the same set of lines, the results of the individual students must give fair curves. In this way each student is familiarized with the use of the integrator and the methods of stability calculations. A great deal of long and laborious work for the individual is saved—work which too often takes edge off the student's enthusiasm.

After the work is finished (and two radically different ships can be assigned with a class of ten or twelve), the class is assembled and the results are fully discussed and explained.

The above is given as a typical illustration. The same method is used for displacement sheets, launching calculations, subdivision and flooding calculations, and to some extent in ship design. The main consideration is to assign the work so that each man must do a certain representative portion independently and check with the others, and together finish a complete calculation as in practice. Long and tedious numerical work on each problem assigned is thus avoided. So far the method has operated well, and a large amount of time has been saved for other purposes.

A third means of increasing time for other subjects is by the elimination of all shop work at the university The student's practical training is gained by summer work in the industry, as will be brought out later on.

Below is given a summary of the course by years:—

The Course in Ship Construction and Marine Transportation.

FRESHMAN YEAR.

First term.

Advanced Algebra (3).
Elementary Chemistry (2).
Chemical Laboratory (2).
Elementary Mechanics (3).
French (3).
or Spanish (3).
or German (3).
Engineering Drawing (3).
Construction (2).
Gymnasium (1).

Second term.

Plane Analytic Geometry (3).
Qualitative Analysis (3).
Stoichiometry (1).
Elementary Mechanics and Heat (3).
Physical Measurements (1).
French (3).
or Spanish (3).
or German (3).
Engineering Drawing (2).
Construction (2).
Gymnasium (1).

Summer Term:—Land and Topographic Surveying.

SOPHOMORE YEAR.

First term.	*Second term.*
Differential Calculus (4).	Solid Analytic Geometry and Integral Calculus (4).
Electricity and Magnetism (3).	Light and Sound (3).
Mechanics and Heat Laboratory (1).	Light, Electricity and Magnetism Laboratory (1).
English (3).	English (3).
French (3).	French (3).
or Spanish (3).	or Spanish (3).
or German (3).	or German (3).
Economics (2).	Naval Architecture (2).
Ship Construction (3).	Accounting (2).
Physical Education (1).	Ship Construction (2).
	Physical Education (1).

Summer Term:—Work in Shipyard on Hull Construction.

JUNIOR YEAR.

First term.	*Second term.*
Strength of Materials (4).	Economic Geography (3).
Strength of Materials Laboratory (1).	Hydraulics (3).
Naval Architecture (3).	Alternating Currents (2).
Dynamos and Motors (2).	Hydraulic Laboratory (1).
Dynamo Laboratory (1).	Dynamo Laboratory (1).
Heat Engineering (3).	Marine Engineering (3).
Business Law (2).	Steam Engineering Laboratory (1).
Astronomy and Navigation (2).	Finance (2).
Physical Education (1).	Machine Design (3).
	Physical Education (1).

Summer Term:—At Sea or in Shipyard Machine Shop.

SENIOR YEAR.

First term.	*Second term.*
Naval Architecture (2).	Metallurgy (3).
Ship Design (3).	Ship Design (4).
Marine Engines and Turbines (4).	Reinforced Concrete (3).
Steam Engineering Laboratory (1).	Contracts and Specifications (2).
Structural Steel Design (4).	Foreign Trade (4).
Foreign Exchange and Marine Insurance (3).	Shipyard Plants and Terminal Facilities (3).
Industrial Management (2).	Physical Education (1).
Physical Education (1).	

It will be observed that the first year is practically no different from other engineering courses and needs no comment.

In the second year a study of ship construction, consisting of class and drawing-room exercises, is begun. This course is given here to familiarize the student with ships and ship construction early in his course and to prepare him for the summer work to follow at the end of the second year.

In the classroom the main features of ship construction are covered, with Holms as a text. In the drawing room each student fairs up a set of lines from partial offsets given him, draws a midship section to the rules of one of the classification societies, and works up an inboard profile construction plan. The midship section is finished completely and traced, but the inboard profile is more in the nature of a study plate and is in no respects a finished drawing. This inboard profile is used to work up various details studied in the classroom. Thus deck construction, pillar and deck-girder arrangement, bow and stern construction are worked out. The idea is to increase the student's knowledge of ship construction by letting him draw up some of the details himself. No attempt is made to complete the drawing as would be done in practice. Thus, for example, the pillar and deck-girder construction is only worked out for one hold and the deck plating only drawn up for part of one deck, the object being to cover as much of a ship's construction as reasonably possible, to drive home the classroom work and arouse the student's interest for his summer's work.

No attempt is made to study the strength of ships in the second year except in a very general way, for the student at this point has not had the proper preparation in strength of materials and applied mechanics. In drawing up these details, merely working from previous practice by use of classification society rules and other drawings, the student learns his limitations and gets a greater incentive to master the theoretical work of the following year in strength of materials.

Naval architecture—displacement, center of buoyancy, stability and trim—is also given in the second year in order to bring the student in touch with his professional work early in the course.

It is very important that the professional studies start early in the course to arouse the student's interest early so that he will have a greater incentive to master the more abstract work of his course.

Following the sophomore year the student is required to spend at least eight weeks in a shipyard on hull construction. Lehigh is probably one of the first universities, outside of those giving cooperative courses, that requires this practical summer work as a requirement for the degree.

The endeavor is made to distribute the men in as many different yards as possible, so that, when they return in the fall and compare notes, a number of different types of ships, methods and practices will be represented.

Although in eight weeks a student will master but little of the art of shipbuilding, yet this practical training introduced in the summer vacation is of incalculable value. First of all it rounds out his sophomore course in ship construction and clears up many little points that may have confused him in his college work;

secondly, he unconsciously acquires an engineering instinct, a knowledge of ships and methods of construction and handling material, and a sense of proportions and dimensions of ships and machinery. Besides this, the student is at an age when he can easily mix with working men and become familiar with their viewpoint—a great benefit to one who in the future may be in charge of men.

A second summer's work of eight weeks is required following the third year. This summer is, if possible, spent at sea. We all recognize the importance of sea training introduced at this point in the student's education. It gives him that knowledge of ships and ship operation that can be acquired absolutely in no other way; yet this knowledge is necessary for him to have the proper appreciative attitude towards the studies of the senior year.

In case we are unable at any time to arrange for all the men to go to sea, work in a machine shop is substituted.

During the third year many of the basic engineering studies begin, practically all supplemented by laboratory exercises. These include strength of materials, hydraulics, heat engineering (thermodynamics), machine design (drawing) and electrical engineering.

Special attention is given to electrical engineering on account of the growing importance of electricity on shipboard for propulsion and auxiliaries.

The course in naval architecture during the third year is given over mostly to the powering and propulsion of ships. Taylor's "Speed and Power of Ships" is used as a text. Vigorous treatment is given to resistance, model tests, ship forms and powering. Both Taylor's and Baker's methods of powering are treated, and power curves are worked up from model-basin resistance curves. The study of propellers takes up both Taylor's and Dyson's methods in detail. All the work in powering and propulsion is illustrated by numerous practical problems.

The junior course in marine engineering deserves special mention. This course, following a theoretical course in thermodynamics and steam engineering, takes up a survey of the whole field of marine engineering and applies the theoretical work of heat engineering to problems in marine engineering. The selection and layout of a ship's power plants, with the relation of the different units to one another, is given special attention. The student is required to carry through calculations for the sizes of the various units of the power plant, emphasis being given to the economical working of the plant as a whole. No attempt is made here to study various types of propelling machinery in detail, that being left until the senior year.

The senior course in marine engineering is given over mostly to a detailed study of propelling machinery and auxiliaries. During the next few years a large proportion of the time devoted to this course will be given over to Diesel engines.

The study of ship design is begun in the fourth year. Here each student carries through a design for assigned conditions, selecting the proper dimensions, coefficients and displacement for assigned speed and deadweight and works up the lines, estimated weights, etc.

It is planned to carry through several typical designs, the students being

assigned by groups to different problems. Thus, as already pointed out, when the more laborious work comes along, such as subdivision and flooding calculation, several men can work together on one design, each one working out certain phases independently, yet together finishing a complete calculation.

No work is repeated that has already been covered; thus stability curves, displacement sheet, launching calculations, propeller design, etc., which have been treated earlier in the course, are not repeated here, although the stability under various conditions of loading is covered. Coefficients and data from good practice are introduced and the student taught to apply these to his design.

A fair amount of attention is devoted to structural details in the senior year, but the main point desired is to have the student grasp the whole problem of ship design. A thorough knowledge of details and methods of construction can only be acquired by actual experience in the shipyard.

Both here and throughout the college course special attention is given to the economic features of ship design and operation. In ship design the layout of the hatches and cargo handling gear is studied for shortening the detention of a ship in port, and the relation of size and speed of a ship for a given trade route receives some attention. A study is made of operating expenses to emphasize the importance of detention on annual profits. In steam engineering, types of machinery and fuel for the most economical performance of the ship in service are given particular emphasis.

A subject of great value introduced in the senior year is a course in structural steel design which takes up a detail study of stresses in structures, and especially those in ships. This course is given by the civil engineering instructing staff in structural design. The work in ship design covers the strength of a ship on a wave and curves of weights, loads, shearing forces and bending moments are worked up.

Attention is especially directed to a course entitled "Shipyard Plants and Terminal Facilities." This course consists largely of lectures by outside men, conferences, inspection trips and papers by the students. During the next few years most of the time will be spent upon terminal facilities, so hopelessly inefficient and out of date in America today. We realize the great importance of this subject and would like to devote more time to it, but at present cannot see our way clear.

The business training given, which is an important element of the Lehigh course, is handled by the College of Business Administration. It starts out in the second year with a course covering the principles of economics followed by a course in accounting. In accounting, the emphasis is more on the reading than the keeping of accounts, and sufficient practice work is given to illustrate the principles studied. In the junior year business law and finance are given, two hours a week throughout the year being devoted to these subjects.

A course in economic geography is given in the third year which treats the physical features and resources of the western hemisphere. Particular attention is given to the present and prospective commercial relations of the United States with other maritime nations.

These subjects given in the second and third years are preparatory for the special courses in marine insurance, foreign exchange and foreign trade given in the fourth year. The courses in marine insurance and foreign exchange and foreign trade cover the following subject-matter:—Money exchanging, financing of exports and imports, factors determining exchange rates, statistical studies in the field of foreign exchange, instruments and forms used in foreign trade; principles of insurance and the peculiarities of marine risks; historical and statistical study of international trade; the organization of steamship lines; combines, export associations and rate agreements; line and charter traffic; steamship ports and the influence of the hinter land; and the relation of inland navigation and railway transportation to ocean commerce.

As a knowledge of modern foreign languages is of great importance to those men who engage later in international trade, the opportunity is offered for study of French, Spanish or German throughout the freshman and sophomore years. The student continues in the freshman year the modern foreign language accepted by the university for entrance. In the sophomore year he may elect French, Spanish or German.

Lehigh is especially well situated to undertake this work in education for the upbuilding of our merchant marine. The offices of the Bethlehem Shipbuilding Corporation are opposite the entrance to our campus, and the shipbuilding and shipping centers of New York and Philadelphia are only a few hours away and can be easily visited by our students on inspection trips.

We are pointing out to our men that their training is not finished at graduation but that it has just begun. We shall urge them, except those who elect to enter the shipping field immediately at graduation, to spend a year at practical work either in a shipyard, engine works, or on shipboard, so as to fully round out the practical side of their training.

The aim of the course, in conclusion, is three-fold:—First, to turn out men who may in the future become specialists in naval architecture or marine engineering; second, to graduate a type of man who, after devoting several years to engineering work along marine lines, will be prepared, when the opportunity arises, to enter executive positions in the shipbuilding and shipping fields; third, to prepare men to enter the business end of shipping directly upon graduation.

DISCUSSION.

ACTING PRESIDENT:—I am sure we are all very much interested in this paper, because I presume that everyone here has taken a course similar to this. The paper is now open for discussion.

Prof. George F. Crouch, *Member:*—This paper by Professor Chapman is of great interest to me personally, and, I imagine, to anyone who, like myself, is engaged in the work of giving instruction in naval architecture and marine engineering. The solution which Professor Chapman has reached of the problem of presenting courses in naval architecture is in many respects strikingly different from and in other respects decidedly similar to, that which we have worked out at the Webb Institute of Naval Architecture.

The planning and carrying on of a course in naval architecture is a good deal like planning and building a ship. The whole matter is more or less a compromise. No one can make a course which is perfect, and I believe no one will claim that any course is perfect. We cannot turn out practical and competent naval architects and marine engineers, we know that. In planning a course we must choose those items which to us seem to offer the best compromise. We try to modify our courses according to the results we get with our graduates.

In taking up this problem a few years ago we recognized at the institute the great importance of the management and executive side of shipbuilding. We were originally devoting our time to what might be called the strictly technical side of this subject. The management and the executive side of the industry was brought to our attention by an alumnus, Mr. McAuliffe, who had some of the problems of the Emergency Fleet Corporation to deal with. I imagine his experience led him to believe that we could do something in the institute towards training our students to appreciate at least that such problems existed.

That necessitated a more or less complete revision of the course as a whole. At the time when we were making the revision, we were taking up the subject of giving our graduates degrees. Unfortunately, the Webb Institute is situated in the State of New York, and the legislature of that state vests the control of the granting of degrees in a Board of Regents. The Board of Regents must follow the laws enacted by the legislature and is compelled by law to demand certain specified and definite requirements for a degree.

The work we are doing at Webb Institute, in spite of the appearance of a contrary statement not long ago, is, I believe, equal in rank to that done in the colleges. It is not preparatory work in any sense, for we go quite far in mathematics, in English and applied mechanics. Our one weak point is the lack of shops. This lack we meet in exactly the same way that has been considered to be an advantage in Lehigh University, and we have met it in that way for over twenty years.

We decided in making this revision, above all things, to keep the practical efficiency of the course unimpaired. The work in management which we have introduced is more or less elementary and is more to give the student an idea of the fundamentals of management and of the executive side of the shipbuilding work than to go into the details of finance and economics.

We have felt in laying out this course that the training required in finance, in business administration and accounting could be obtained quite satisfactorily by the men before going into those branches, in some of the business courses which are offered today. We have introduced a series of graduate lectures on many of these topics and also instituted a series of visits to plants where these points are taken up.

A second point about the degree is that we found it impracticable in connection with a complete course in marine engineering and naval architecture. We could have dropped out the marine engineering, or made it more or less general, and retained the naval architecture, and given the biology and other things that the regents wanted us to

give; or we could have kept the marine engineering and left out some naval architecture, but to put in all the things they wanted, many of which seemed absolutely unessential, did not appeal to us. The regents themselves stated that it would doubtless hamper the efficiency of our work as a school in naval architecture to attempt to give a degree with the laws of the state as they are at present framed.

We have tried to keep our course as full as we can on the technical side. In comparing our course with that at Lehigh University, I find the chief difference in the Lehigh course is in the amount of detail on the marine engineering side and the substitution for those details of courses in finance, economics and management. Evidently the aim of the course at Lehigh is somewhat different from the aim of our course. If things work out right at Lehigh, perhaps the executive in charge of work may be a graduate of the Lehigh course and he may employ one of our men to do some of the marine engineering.

I have been glad to see recognition given by Professor Chapman to the fact that it is impossible for any one student to take up and carry out in the four years the complete design of a ship and make all the calculations for it. Those of us who have worked in drawing offices know how long it takes to make a general arrangement plan, an inboard profile and outboard profile, line drawing, deck and shell plating plans, to make the model and lay off plating and make the necessary calculations carrying out the details of the design. I like to see recognition of that limit. No one student, doing all the other things he has to do, can accomplish more than a finished draughtsman would attempt.

With regard to summer work, we have eight weeks of it, not as a requirement for a degree, as we do not grant degrees, but each summer we have required this work ever since our courses at the institute were started. We find it one of the best things we have done. It tends to make the student alert and leads him on in his studies—he gets in touch with practical conditions. Turning a cylindrical bar as a class exercise in a university shop does not compare in efficiency in the education of a shipbuilder and marine engineer with his going on board a ship, working as a ship fitter's helper or working under commercial conditions in a machine shop.

This course at Lehigh is a beautifully balanced course, and I think we can compliment Professor Chapman on the way he has planned it.

Mr. Harold F. Norton, *Member:*—There is one point I wish to bring out in connection with this paper. I remember very well when I was studying in the course of naval architecture at Cornell a number of years ago that, having had considerable shop experience before going to the university, the thing that always puzzled me was how a ship was built with the knowledge we gained at the university. We used to draw up a set of lines, a midship section, and an inboard profile, and I used to say, "A ship surely cannot be built with these plans only. How do you build the rest of the ship?" They said: "Oh, you have to make a lot of plans, of course, of all the details of the ship construction." I then asked: "Where are the plans?" The reply was: "We do not happen to have any."

It has always been remarkable to me that the universities did not possess themselves of a complete set of plans of at least one type of ship to show the student. If he could not draw all the plans, at least he could see them all and could appreciate how a ship is really built. It has been astonishing to me in all the years I have been at Newport News that there have been so few applications from colleges and universities throughout the

country for plans of a merchant ship. I would like to say that if there is any college or university requiring such a set of plans, if they will apply to us at Newport News we shall be delighted to give it to them. I personally believe that a set of plans of that character, every one of the necessary plans to build a ship, will be of more use to the student in naval architecture than half of the lectures he receives.

I would like to disagree a little with the idea that working in the shop and turning a bar on a machine does not teach a man how to build a ship. The fault we find in connection with the university students is that they are not sufficiently familiar with the details of construction, and by these details of construction I mean not merely the fitter's work. We want a man who knows how to make a casting, and who knows what a pattern ought to be like and how the machine details ought to be worked out. He is quite as useful to us on the hull side of shipbuilding as the man who has never seen anything but fitter's work.

In our apprentice system at Newport News, the one thing I have fought for all along is that our apprentices who expect to complete their course as hull draughtsmen shall not be confined in the yard merely to work in the ship-fitters' department, but that they shall go into the machine shop and foundry, the pattern shop and carpenter shop, the joiner department, and into any other department where they become familiar with ship details. There is no doubt that these men will learn the ship-fitting part in connection with their own work. It seems to me that one thing the universities and colleges should teach a young man is that he need not worry so much about becoming a splendid executive. We will endeavor to attend to that after we get him, if they will only properly start the man and give him knowledge enough of ship construction and ship details to be of use when we first get him. We will not worry particularly about the necessity of teaching him some of the necessary calculations to carry on his work, or about his absorbing the necessary details for advancement; if he possess suitable material for an executive, we shall certainly endeavor to develop those qualities in him.

Mr. Walter M. McFarland, *Honorary Vice-President:*—As one of the Committee on Papers, I was much pleased when this paper by Professor Chapman was submitted, not so much on account of the specific statements made in it, although the paper is an excellent one, but because it would bring the subject up here where we have so many practical men, so called, from the shipbuilding and allied industries, and would give them a good chance to discuss the work of technical schools and say wherein they think the courses at such schools are good or else are deficient.

I have taken a great interest in technical education all my adult life. It happened not many years after I left the Naval Academy that I was assigned as an assistant professor at Cornell University and spent a couple of years there, so I got some knowledge of the professorial attitude; and in recent years I have had the privilege of being one of the trustees of Webb Institute, where I have given a great deal of attention to the matter. The thing which has always impressed me for many years has been, as stated both in the paper and by Professor Crouch, that the time at the school is so limited, and there is so much to teach that the course must be a compromise. It seems to me that the proper attitude for the shops or the shipyards, to which the boys go afterwards, should be to recognize that the school has done its part if it has given them a thorough training in fundamentals on which they can build when they go out into practical work.

Many of you probably remember the time—at least the older ones do—when the

technical schools really began, and that there was a demand on the part of the shipyards and factories that a graduate of one of these technical schools should be able practically to take the position of general manager. In other words, they seemed to flatter the schools with the idea that you could take a boy of sixteen or seventeen years of age and give him four years of training, which would take the place of twenty-five or thirty years of actual experience in business affairs. That, of course, is ridiculous. We all know, perhaps, that the greatest element of value of the leading men in all the shipyards is this very experience. If the schools do give this thorough grounding in fundamentals, then I think they have done their part and all that can be asked of them.

Then, of course, the question is: — What are the fundamentals which will be given? A paper like this of Professor Chapman brings the matter up and erects a target at which anyone can shoot if he wants to.

I was glad, indeed, to hear Professor Crouch's remarks, and I wish to confirm his statement that we at Webb Institute had already thought of some of these ideas that Professor Chapman has brought out in his paper, and that we are trying to graduate our boys with this start to enable them to develop into executives as well as into draughtsmen or technical men only. As Professor Crouch said, Mr. McAuliffe, one of the Webb graduates, had a chance to see from his own personal experience that if the boys who come from technical schools had been given some elementary training in management, in executive work, they would be in a better position when they got out into the shops to develop themselves for filling such positions. We all know it is commonly said that every large concern is constantly on the lookout for good men to advance, and that, of course, is true, but if boys never hear anything about management at all, the tendency for them is to start in minor positions without thinking of this thing unless they are unusually ambitious. They do not have as broad an outlook as they otherwise would have, and I believe their outlook should be broadened to make them more useful men in the shipyards where they go to work.

Mr. P. J. McAuliffe, *Member:*—As a graduate of Webb's Academy and a member of its Board of Trustees, I want to take this excellent opportunity to thank the naval architects and marine engineers for their very much appreciated assistance in placing the students of Webb's Academy during the summer months in the various shipyards.

I also want to say that while I was in the Shipping Board we endeavored to collect a great deal of information concerning plans, something like what Mr. Norton spoke of, to be sent around to the various schools of naval architecture, and if any school of naval architecture, marine engineering or shipbuilding has not received these plans, they can probably get them from the Shipping Board, as these plans were to be placed at the disposal of any technical institution in the country.

Prof. H. C. Sadler, *Member of Council:*—The question of the education of the engineer is one that is open to more criticism and discussion than anything else, particularly at the present time. You know, I think, that there are more fashions in engineering education than there are in clothes. We go through a cycle of fashions and very often go back to the old fashions of thirty, forty or fifty years ago; at any rate that seems to be the tendency at the present time.

I might tell you of an interesting experience we had some years ago. We are always trying to look ahead in Michigan and so decided we would have a meeting of our alumni.

We inaugurated a discussion on engineering education with a view to getting suggestions from them as to what should be given in an engineering course. There were altogether some forty to fifty speakers, and each man had a suggestion for a particular course in his particular line of work, with the result, if we had included all the suggestions offered by our alumni, our course would have been a ten-year engineering course.

I do not mention that in any critical sense, because we got some valuable suggestions, but there is a tendency, I think, after men have been out in practice for some time and have settled into a particular branch of their profession, or in a particular niche, or in a particular concern, to feel if they only had a course in that subject while in the university they would have got ahead so much quicker. Gentlemen, that is a fetish—an absolute fetish. (Applause.)

You never can tell, or a student never can tell, in his four years' course in the university, where he is going to land, and nine men out of ten land in a very different place from where they thought they would originally.

The tendency today perhaps has been more towards the training of men for the executive end. I think that is very important, but the pendulum is apt to swing too far in that direction. We would all like to train as presidents of companies. I think that would be an ideal job, but, unfortunately, probably 75 per cent or more of the men we turn out are going to be the rank and file of the profession, or reach fairly prominent positions, perhaps, so that although I think we ought to bring before students certain features of an executive character, I have a feeling we do not want to emphasize them too much. I am afraid that after a man has taken a course in business management, or a course of training for president of a company, he might be disappointed if on graduation he finds that the companies do not grab him up and make him a president of the company after he has been out of college three or four weeks. There is a tendency for the men to become discontented under those circumstances.

We have adopted a course at Ann Arbor very similar to the Lehigh scheme, with this exception, that instead of laying down rather definite courses for a man to take we have a certain number of hours in the course which are left entirely optional—practically more than half his work in the senior year. A senior student is allowed to select work from any place all over the university. If his bent is towards the sales end of his profession, he can take a course which will help him in that. If his bent is in the managerial end, he can take some work in that. If he wishes to become a real engineer, he can take some more engineering, and we have a feeling if we have given a man a thoroughly good, broad, engineering training first, with the fundamentals of naval architecture and marine engineering, and then leave him free to elect, if he wishes, these other subjects, that we have come, as nearly as possible, to the education of a man for his future life.

As I say, there are many differences of opinion as to how a man should be trained, but I think, fundamentally, we all have the same ideas. After all, the true purpose of the technical school and the university is to give a man the fundamentals of his profession, so that later on in life, when he gets to a position of authority, he is not afraid to tackle new problems. I think it is a mistake to overload a course too much in specialized subjects.

Mr. Charles F. Bailey, *Member of Council:*—I think the Society is greatly indebted to Professor Chapman for his paper and to the gentlemen who have preceded me for their discussion of it.

There are two or three thoughts which I wish to present to the meeting in connection with this subject. Professor Crouch has spoken of the practical training which they insist upon at Webb Institute during the summer months, which training consists of about eight weeks of work in the shipyards. In my opinion this summer work is most important. A great many of the young men whom we take at the Newport News Shipbuilding and Dry Dock Company come from engineering schools, and we depend largely upon these schools for recruiting our technical forces. At Newport News we very seldom go outside of the company for a foreman or an official, and I think that this practice is becoming more and more a custom at all of the shipyards, the policy being to take in young men and let them grow up in the firm's employ, encouraging them to feel that they have a future with the company. So we take young men from the universities and from our apprenticeship courses and develop them and look to them to become the future shipbuilders at Newport News.

We sometimes find that the young men who come to us from the universities expect too rapid promotion, as has been commented upon by Dr. Sadler and others. I hope that you gentlemen who are teaching naval architecture and marine engineering will impress upon these young men that it is a long road in shipbuilding to position and responsibility and that the money consideration which young men receive during their early years in this work may not be as great as in some other business or profession, but that the work is full of interest and opportunities for useful service and gradual advancement. The money which they may earn as young men is only a small portion of their remuneration.

Along this line I want to refer to a remark made by our chairman several months ago in ordinary conversation, which would be of great benefit to all young men if it were indelibly impressed upon them, in order that they might see that one's best continued effort is the only way to success. As our chairman put it:—"No man can live on his reputation; a man lives by making a reputation."

Mr. William T. Bonner, *Member:*—While we are discussing colleges I would like to mention Drexel Institute and the Franklin Institute of Philadelphia, two schools that have introduced practical courses in various branches of the shipbuilding industry. I can speak a little more familiarly about Drexel, as I happen to be on the Joint Committee which was appointed to arrange a course for welding at the institute, but which is not yet in full operation. However, we are getting on very well.

Drexel Institute originally started out as a trade school but has been getting away from that idea, and the aim now is to devote attention, as far as possible, to higher education in the different practical branches which lead to the shipyards, to the railroad shops and to other institutions of that character, and thereby assist, in a complementary way, in the development of the productive side of the mechanic.

We are not setting out to develop only presidents and managers but hope to produce a class of foremen and superintendents who will not only understand the practical side of the different trades at which they are working but how they may get on and become managers of the men. We are also trying to instill in the minds of the students the necessity of using their brains and their consciences, as well as their hands, in carrying on their work.

I probably never felt so hurt in my life as I did a few days ago, while inspecting some welding work that was going on, and the leading welder, who was rated to be one of

the very best mechanics of that line, showed his hand. I had criticised part of the work which he was doing, and he explained that his reason for not doing it as he knew it should be done was because he could not "make his time." By this he meant he could not accomplish his speed bonus by doing the job in the manner he believed best suited to insure first-class workmanship, therefore he did not hesitate about cutting across corners even at the risk of sacrificing quality, and with it a successful job.

While I did not altogether agree with the welder's theory as to how the work should be done, I considered that an insignificant matter as compared with the principle involved in his admission that he was obliged to use inefficient methods in order to save his bonus.

Now, gentlemen, I think it is up to you as executives and administrators to take a hand in this education and try to impress your assistants and your foremen, and through them the workmen, that it is not merely a question of how many dollars or how many cents they are going to get out of a job, but how well they can do the work. I believe the colleges should feature that in their courses, as being just as essential as knowing how to work out formulas for calculating sizes and estimating costs.

I would like to take exception to the idea our chairman expressed that the principal function of welding is its use as a corrective art, and to have you feature welding in your minds as a productive and a constructive art. It is all right to use welding for corrective work, as far as it need be so used, but let us hold our men to the same strict accountability—pardon the expression—for avoiding the necessity of corrections as they were admonished to use prior to its introduction, and let us also feel that we are equally justified in developing welding for protective and constructive work.

Mr. Stevenson Taylor, *Past President:*—As president of Webb's Academy and Home for Shipbuilders or Webb Institute of Naval Architecture, as it is now known, I wish to say a few words.

Most of you, perhaps, know this institute is quite different from any other institution of the kind in the whole world. It was founded by the great shipbuilder, Mr. William H. Webb, in 1889. He made me one of the charter members of the Board of Trustees. Upon his death in 1899, I became his successor as president. The institution was called by him Webb's Academy and Home for Shipbuilders. It is divided, therefore, into an institution for instruction and an institution as a home for indigent shipbuilders. Its foundation is practically settled, so that I do not imagine that it would be attractive for those who desire to leave large sums of money to institutions of that sort. It is probable that we shall not increase its present funds to any considerable extent, and therefore we are limited as to the number of each class of beneficiaries.

We give an education to young men in naval architecture and marine engineering, confining most all of our efforts to those lines, and I am proud to say that during its comparatively brief existence the success of the graduates, considering the size of the school, has been such that we need not be ashamed of its progress or its work. During the recent demand for shipbuilders and people with some knowledge of naval architecture and marine engineering, a great many of our graduates—one hundred and forty out of a total of one hundred and ninety-two—served the United States Government in its various departments, doing their best and receiving high commendation.

Limited as it is by its endowment, which will probably not increase, we must go along the lines we have followed heretofore. It probably would be advantageous to add

courses to the institute, but after all, speaking from my own personal experience in my own business and my experience in the work of the institute, I think it is just as well to give a young man a fundamental education that will be of value to him in the future, leaving something to his own individual initiative. I believe that our success in life depends upon ourselves, and you may put some young men through the universities of Michigan, Cornell and Lehigh, the Massachusetts Institute of Technology and Webb Institute—all of them, one after another—and they will never make a success.

I would like to say to the young students now present that their success in life depends upon themselves more than upon any institution which is trying to help them.

Webb Institute is distinctive in this respect—it gives to the student not only his education, but it furnishes him with all books and materials necessary for the particular education he receives. It also furnishes him with board and lodging, the only physical requirements being that he must have good health and necessary clothing. We are limited in respect to admissions to the institute, because we can only admit students from that class of young men who cannot, through means of their own or of their parents or guardians, procure education in other places like Stevens, University of Michigan or Cornell or the Massachusetts Institute of Technology, so that we have a distinct class of students. I sometimes think, concerning the education we give these young men, who thus get everything for nothing, that it is an advantage for them that they have no great resources of their own, because that condition, aided by the special education, rather increases initiative.

I have been very much pleased, Mr. Chairman, with the discussion of this subject, and I must confess I was a little surprised, for I did not realize how interesting the subject is to so many members of this Society.

In conclusion I make application now to Mr. Norton for a full set of drawings of one of his ships, and also to Mr. Bailey for a set of his drawings of machinery.

THE ACTING PRESIDENT:—Is there any further discussion?

CAPTAIN JOSEPH H. LINNARD, C. C., U. S. N., *Member of Council*:—A question has arisen in my mind, apropos of what has been said, and that is that the Webb Institute is favorably known to all of us who have young men come to us for investigation as to their qualifications for shipbuilding in various directions, and I often think that perhaps one of the reasons for success in the Webb Academy has been the fact that these men who go into it are selected from the beginning. They cannot get in unless they have certain qualifications to start with; they are a select body before they start. We see so often young men who enter into institutions with the vaguest ideas as to what they desire to learn and what they may expect in the future. Whether they have natural aptitude for the profession or occupation they propose to adopt is frequently a minor or non-existent question, and I have wondered, in listening to this discussion, whether some of these other institutions make any endeavor, before entering young men in their classes, to find out if they have any natural aptitude for the profession. It would seem to me to be one of those fundamentals which has been neglected in the past.

MR. STEVENSON TAYLOR:—The question raised by Captain Linnard is quite important, and I may say just a few words about it. We hold examinations in September, and the examinations are very stiff. We insist that a high school education is necessary

before undertaking the further study of naval architecture and marine engineering. Then we have very severe examinations twice a year, with the result that we take in from fourteen to eighteen students in September, and we graduate perhaps six, eight or ten, and I think we have gone as high as twelve at the end of the fourth year.

The officers of Webb Institute feel that, where students receive so much at no cost to themselves, the institute is entitled to the best men of their kind and that they in return must do their best; otherwise, we might, due to our limit in number of students, be keeping in uselessly an impossible student, thus keeping a good one out.

I should have also told you why we changed the name from Webb's Academy and Home for Shipbuilders to Webb Institute of Naval Architecture. When Mr. Webb founded the institution, the word "academy" meant to him as much as the word "university" now means to most people. You will recall that since 1889 a number of colleges have been changed to universities, and a number of high schools have been changed to colleges, and the word "academy" is now defined so specifically by the Regents of the State of New York that it means nothing more than a high school. Therefore the Board of Trustees—and I may give here special credit to our vice-president and chairman of the school board of our institution, Mr. McFarland, for advocating the change—thought it was wise to change the name for the benefit of our students, because the graduate of an academy does not now seem to mean much, whereas a graduate of an institute seems to carry more efficiency.

Mr. H. P. Frear, *Member of Council:*—The outline of the course in naval architecture at Lehigh University as given by Professor Chapman is a valuable contribution to the proceedings of this Society and introduces innovations which should broaden the field to students who inherit both engineering and executive ability.

Comparatively few who graduate from technical schools possess these combined qualities, and some possess neither of them. Bright scholars or students, of which there are many, have little difficulty in passing entrance examinations or securing their degree, but scholarship alone is not the only requisite for success in a particular line. Being a college man myself, I am a believer in technical education, but during my thirty-seven or more years' experience I have met with many cases of graduates who had missed their calling and could not compete with men of ordinary ability, trained from the ranks, but having in some degree a natural instinct or bent for their work.

Some boys have a decided mechanical inclination and demonstrate this from childhood by playing with mechanical toys and accumulating tools or apparatus with which they either make things or experiment. Many have come to me for advice before deciding to enter a technical school, and I have never failed to recommend a college education to those apparently having inherent mechanical ability. A large number, however, had not found themselves and possessed a more or less vague idea what they were going into. If there were some process by which boys of the latter class could determine their fitness for the profession before entering college, many mistakes would be avoided.

With this in view it has been my invariable practice to recommend to every young man who came to me for advice, to first go into the shop for one or two years; after which I was able to advise with greater confidence. The shop not only helped the doubtful boys to decide before going too far whether or not they had made a wise choice, but it taught them all how to work and be useful, which was more than they sometimes learned in college. I always had some kind of a shop of my own as long ago as I can remember,

made model yachts and sailed them and turned many of the tops and baseball bats used in the locality, and there was never any question regarding the line of work I would follow. Nevertheless, on advice, I worked for two years in the Honolulu Iron Works before going to college and consider them among the most valuable years of my life.

Summer work in shipyards and vacation sea service, as provided in the Lehigh course of study, cannot be too highly commended and may be of more value than shop work during the regular course, but I believe that the majority of the students would receive greater benefit and have greater earning power at the start if the time devoted to economics and business administration subjects were devoted to shop work, reserving to a later date the study of these special subjects described by the author, for only those who would be most benefited.

To sum up, I doubt if the course would be of as much value to the average boy as would be the case if the usual four years' course was taken to be succeeded by a fifth year devoted to the subjects appropriate to an understanding of the business part of shipbuilding and shipping by those who especially desire to take it. My main objection is not that the subjects selected are not of value but that to crowd the ordinary four years, considered none too much for a technical training in naval architecture, with these other studies, may result in a student acquiring a false impression that on account of these special studies he would be capable of stepping immediately into an executive position requiring special aptitude and often years of actual experience. On the other hand, a boy goes to school in one sense for an education and, from my observation, the particular school he goes to or the character of the course amounts to less than the characteristics he may possess. If Mr. Chapman could select only those who would be most benefited, his course of study would be ideal.

Prof. C. H. Peabody, *Member of Council:*—I wish to congratulate one of our alumni on being able to start a new course in a fresh field. Those of us who have labored many years in this field have had all the conditions presented this morning brought to us many times, and we find in many cases we cannot at once change a large and somewhat inflexible institute to meet the conditions which are suggested to us. It may, perhaps, take us a little longer to accomplish such a result.

I wish, however, to say that it is a great comfort to me, now that I have finished with this rather difficult task, to be told by a number of men in practice that a good, sound training is the best thing. That is what we thought we could give them and what we have attempted to give them, whether they acquire some of these other accomplishments or not.

I am somewhat doubtful about the courses in the business management and the production of executives. The whole thing is rather new and is to be justified by its works. I hope that it will be.

As for the matter of having sets of drawings, I personally have worried all of the people I know well enough, asking them for sets of these drawings, until I have become a nuisance to them. Our collection of drawings cannot be compared with the drawings which are found in shipyards, but we have, at any rate, a number of sets which are as complete as the shipbuilders would be willing to let us have, and which I think are quite sufficient for our purposes.

Mr. John Flodin, *Associate Member* (Communicated):—Professor Chapman is to

be congratulated, not merely on his valuable contribution to the Society but also, and chiefly, on the excellent curriculum he has worked out for Lehigh University.

The establishment of the Lehigh course along the lines set forth by Professor Chapman is all the more pleasing to me because for many years it has been my belief that a departure of this nature was very much needed. This belief, and the general character of the education that I thought would best fulfill the requirements of shipbuilders and operators, I endeavored to set down in an article that was published in Marine Engineering about a year ago. My conclusions were, however, that even for a course of much less technical nature than that offered at Lehigh, a four-year period is not sufficient. Professor Chapman has succeeded in drafting out a four-year program that is considerably richer in technical material and that, furthermore, contains some very useful courses in economics, languages, etc., which results in the rather heavy study schedule of about twenty-six hours per week, average, for the freshman and sophomore years, and about twenty hours per week during the junior and senior years. This, I believe, is rather more ground than other technical colleges have been able to cover satisfactorily. But if this program can be carried out without injuring the health of the students—I know that Lehigh University will not permit a lowering of its standards—Professor Chapman's efforts are surely worthy of being crowned with every success.

An especially fortunate feature of the Lehigh program is the requirement that the students spend their summers in shipyards or at sea. It seems, however, that the time allotted for this part of the training is not sufficient and that it might be found advantageous to substitute additional practical work in shop or at sea for the summer surveying course now required after the freshman year.

ACTING PRESIDENT:—Has Professor Chapman any reply to make to the discussion?

PROFESSOR CHAPMAN:—I am indeed gratified at the discussion that my paper has brought forth. This course at Lehigh was started as a departure in a new field. Apparently, most of the speakers do not realize the fact that the aim of the course is not to educate men for shipbuilding alone. We are also trying to prepare young men to enter the fields of shipping and ship operation; and we believe that anyone entering these fields should have a thorough training in the engineering aspects of shipbuilding, naval architecture and marine engineering.

I think we are all agreed that what is wanted in any engineering course is a thorough grounding in fundamentals. That is what we are attempting to do in this course. We have dropped out some of the courses that we think are too specialized, and in place of them we have added fundamentals in other fields. Take, for instance, the course in accounting. Many men who have graduated from our technical schools in recent years know nothing at all of the principles of accounting and finance. What we are endeavoring to do is to teach the men the fundamentals of accounting and a few of the principles underlying finance and business management, so that, if the opportunity of a managerial position comes to them, they will at least feel that they have some ground work in these business subjects. If they have the proper preparation to start with, they can, in the years of training that follow graduation, build on that foundation by outside study and reading.

We are endeavoring, as I brought out in the paper, to drive home to these men that when they finish their college course they have just started their period of training.

We have some good friends at the Bethlehem Shipbuilding Corporation, just across from our campus, that we are going to ask to keep on preaching the same thing to them; so that by the time these men leave school they will realize that there is no large opportunity for them unless they first go into practical work of some kind, either in a shipyard or on board ship. When it comes to graduation, I feel confident almost every one will either go into a shipyard or on shipboard and start right at the bottom.

There was some criticism made by Mr. Crouch of the lack of marine engineering. I wish to call his attention to the fact that we have a course in steam engineering, a course in steam engineering laboratory and a course in machine design given by our department of mechanical engineering. There are also two courses, a junior and a senior course, in marine engineering. We do not teach marine engine design, but we teach the fundamentals of marine engineering and go over the field of marine engineering as a whole to give the students a clear understanding of present day practice. Then we take up the propelling engine and auxiliaries one by one in a detailed study, so that the student has a pretty comprehensive idea of the ship's power plant. He will, however, not know much about engine design except the underlying principles. We believe, if we can get the men well grounded in the principles of strength of materials and mechanics, that any of them who want to follow marine engineering will be able to learn the details of design in the industry.

Mr. Flodin, in his written discussion, offered some criticism of the fact that we had a pretty stiff schedule. Our hours, nineteen to twenty a week throughout the year, are the same as in the other engineering courses at Lehigh, and practically the same as in other engineering schools in this country. While the course looks heavy in outline form, it is no heavier, as just pointed out, than other engineering courses in this country.

Captain J. H. Linnard, C. C., U. S. N., took the chair.

THE CHAIRMAN:—We thank Mr. Chapman for this paper which has occasioned so much interesting discussion. The next business is paper No. 2, "Launching of Ships in Restricted Waters," by Lieut. Commander Harold E. Saunders, Construction Corps, U. S. N., Member. I understand that Mr. Saunders is absent, and I ask Mr. William Gatewood to present the paper.

MR. WILLIAM GATEWOOD, *Member*:—Gentlemen, in the absence of Commander Saunders, I have been requested to read a synopsis of his paper, and the reading of the paper will be followed by the showing of the motion pictures which were taken during the launching.

Mr. Gatewood then presented the paper, after which the motion pictures were shown.

LAUNCHING OF SHIPS IN RESTRICTED WATERS.

By Lieutenant Commander H. E. Saunders, Construction Corps, U. S. N., Member.

[Read at the twenty-eighth general meeting of the Society of Naval Architects and Marine Engineers, held in New York, November 11 and 12, 1920.]

At the last meeting of this Society there was presented by the writer a paper dealing with methods employed to check the speed of vessels during launching, with particular reference to a type of friction launching brake developed and used at the Mare Island Navy Yard. A brief description of the launching and checking arrangements for the battleship California were given, together with a description of certain model experiments conducted in connection with this work. At the time this paper was submitted, however, the launch of the California had not yet taken place and it was proposed to submit a second paper, in the present form, embodying a description of the launch and an analysis of the data obtained at that time.

Before proceeding to a detailed account of the launch, which took place on November 20, 1919, it may be well to note the principal characteristics of this vessel, as bearing upon the data set forth in the following paragraphs:—

Length, over all	624 feet.
Breadth, over all	97 feet 5¾ inches.
Mean draught at launching	14 feet 9¼ inches.
Launching weight complete, with cradle	14,610.5 tons.
Pressure on ways, per square foot	2.4 tons.
Mean declivity of ways	11/16 inch per foot.
Maximum velocity attained	24.7 feet per second.
Velocity when brakes were applied	21.5 feet per second.

In order to obtain data for a complete analysis of the launching, every effort was made to provide sufficient recording apparatus for all phases of the operation. In addition to the standard drum chronograph previously employed at this yard, a second "direct-reading" chronograph was fitted and both machines connected to the ship by wires which were laid out over the ground ways. The arrangement of the latter chronograph, whereby the special "clock," the chronograph dials, the two pressure gauges of the No. 5 brakes and the tachometer (indicating feet per second velocity) were all included in the field of view of the motion-picture camera, is very similar to that shown on Plates 10 and 11 of last year's paper. Plate 2 is a view of the entire brake control station at the head of the ground ways, port side. Plates, 3, 4, 5, 6 and 7 have been included in this paper to supplement the plan, Plate 14, of last year's paper. These illustrations show, in considerable detail, the arrangement of hand pumps, brakes, chains and friction cables as actually installed.

No attempt will be made to include in this paper any details of the launch other than those having to do with the braking operations. The complete launching report, Mare Island Plan No. 24389, has been issued and is now available for the information of those who may desire it. The following paragraphs are copied verbatim from this report:—

OPERATION OF HYDRAULIC LAUNCHING BRAKES.

"Before proceeding to a description of the behavior of these brakes, it will be of interest to note briefly the organization of the special brake detail and the instructions issued for the guidance of the members of this detail. It has already been stated that there were three full rehearsals of these operations prior to the launch and that all the men were given an opportunity to watch the performance of the launching model, when specially rigged with cables and chains.

"The two groups of men operating the hand pumps were directly in charge of a supervisor on each side, who was to superintend the operation and the emergency release of the brakes on that side. It was conceivable that, in the event of a wire cable stranding or kinking, or part of the brake gear carrying away, it might be necessary to release the pressure in one or more sets of cylinders to avoid serious damage.

"The brakes were to be set as soon as their respective cables began to pull through (for the brakes Nos. 3, 4 and 5, port and starboard, this would occur at about 568 feet travel). As an additional signal, two large klaxons were to be sounded by the brake control officer at the head of the slip, when the ship had run the proper distance and the chain cables had straightened out. From the rate of pressure curve, previously determined at the launching of the Zane, it appeared that about eleven seconds would be required to apply the normal pressure of 850 pounds per square inch, and that in this time at least 200 feet of wire would be drawn through the brakes. The initial pressure was set at 50 pounds in order to insure that there was water in each system and that the hydraulic rams would respond at the first stroke of the pump. The water service was turned on each set of brakes at the moment that the vessel was released.

"In case the speed of the vessel had not been sufficiently checked at 1,000 feet travel (under normal conditions the vessel should have stopped dead in the water at about 960 feet travel), emergency pressure, 1,150 pounds per square inch, was to be applied, the signal for this pressure to have been five or six short blasts of the klaxons. When the vessel had stopped dead in the water, the brakes were to be quickly released upon signal of the klaxons, to prevent the strain on the cables from drawing the vessel back toward the slip. Competent observers were stationed near each set of brakes, to note any unusual behavior of the apparatus and to watch the performance of the brake crews. Arrangements were made to take motion-picture observations of two extra gauges on the port and starboard No. 5 brakes. The actual pressure, distance and time readings are shown on Plate 10.

"The chain cables attaching the friction wires to the ship were flaked down

on the floor of the slip so that each chain was clear of the others and separated from its own parts. The tumbling shores, of course, had to remain until the vessel was released, and, although they were directly in the path of the chains, there was no interference and very few of the timbers were thrown about.

"From the moment that the vessel was released until she reached 648 feet travel, everything functioned exactly as was planned. At about 560 feet travel the chain cables began to straighten out, and at about 570 feet travel the klaxon signal was given to apply normal pressure on Nos. 3, 4 and 5. Within less than one-half second the operators began pumping on these brakes, and the first application of pressure was sufficient to straighten out the chain cables. The brakes smoked considerably from the start, due to the fact that there was a protective coating on the wire cables of some fish oil preparation, which burned off on the outside of the strands as soon as the cables began to slide. At about 620 feet travel the cables straightened out on No. 1 and No. 2 brakes, and normal pressure was applied by the operators.

"At 648 feet travel, after the wires on No. 3 port brake had gone about 80 feet through the blocks and a pressure of 500 to 600 pounds per square inch had been applied to the rams, the chain cable on that brake carried away several feet from the pad on the ship's side. The vessel by this time had left the ways and was traveling at a velocity of about 19½ feet per second. At 660 feet travel, the chain cable on No. 5 port brake carried away about 420 feet from the ship's side. The wire cables and the cable reels, traveling at high speed when the wires suddenly stopped in the brake, possessed enough momentum to throw the wires up in kinks behind the brake and to unwind several turns from the reels. Although the friction brakes on the cable reels were not supposed to care for such a sudden stoppage, they should properly have been set to exert considerably more resistance (see Plate 8).

"One by one, as the pressure on the brakes was increased, their respective cables parted, about half of the chains remaining intact when normal pressure was obtained. At 776 feet travel, all chains except No. 1 port had parted; at 780 feet, one of the links connecting the outside wire on No. 1 carried away and the inside wire continued to pull through the brake. Although no further signal was given at this time from the control station, the supervisor in charge on the port side motioned to the operator at No. 1 to hold the pressure on his brake. Mistaking this for a signal to apply emergency pressure, he immediately began pumping to about 1,100 pounds per square inch. At 902 feet travel, as soon as this pressure had been applied, this chain cable also parted and the vessel was then running freely with the exception of the pieces of chain dragging along from the pad-eyes. The starboard and port anchors were then dropped and the vessel brought to rest as described elsewhere in this report.

"Port No. 4 brake was dismantled and opened for inspection immediately after the launching. The wires had traveled about 100 feet through the blocks, but there was no rifling and comparatively little scoring of the grooves. The leading end of the grooves showed practically no wear; the trailing end showed

discoloration on the tops and bottoms of the grooves about half an inch wide and 20 to 24 inches long. The concentration of pressure and the resultant generation of heat were almost sufficient to make the metal soft enough to flow. The same condition on the other brakes indicate that the water-cooling features of the brake must be improved somewhat and that the cooling must be most efficient where the greatest amount of heat is generated. The operation of all other parts of the brakes left nothing to be desired. There was no serious twisting of any of the cables; for instance, one wire of No. 1 port brake slid for about 280 feet in its groove with no such effects whatever. Flats were worn on the outside strands and there was some discoloration of the metal, but the cables, after launching, were in perfect condition for use on the next battleship.

"Six of the broken links from the ten chain cables were recovered, and every one showed either faulty weld or very inferior structure. On one link the weld was perhaps less than 10 per cent effective, and on another there were crystals of metal three-eighths inch square. It is not likely that any one of these six cables pulled even its normal load of 30 tons, to say nothing of withstanding its proper breaking load of 83 tons.

"The fact that two of the cables broke before 600 pounds pressure had been applied to the brakes proves that neither excess load nor sudden application of load was responsible for the failure of the chains in this instance. The motion-picture records show clearly that the cables remained fairly taut and that there was almost no surging for the time that they remained intact. There was no apparatus at Mare Island with which to make physical test of these chains, and it is quite probable that even such tests, carried only to the proof strength of the chain, might not have disclosed serious defects. Chain cables for this purpose are considered treacherous and unreliable and far inferior, except for ease in flaking down, to wire ropes with suitable end fastenings. They have not been used in the past at this yard, and they will not again be used in the future.

"Although in this case the absolute failure of these chains to stand up to their work introduced some unforeseen complications, it is considered that the results, taken all in all, most fully justified the modifications which had been made in the original design of these friction brakes. It is interesting to note from the curves that, had the cables held and exerted their proper pull, the vessel would have been stopped in almost exactly the same distance as had previously been calculated."

An examination of Plate 9 will indicate that some of the chain cables in question carried away before the defective links had suffered any deformation whatever. As an extreme example, it may be seen that the weld on the end link of No. 1 port cable opened out as if it had been simply bent to shape and never welded at all.

Aside from the question of chain cables, the functioning of the hydraulic brakes was, on the whole, so satisfactory that they will be retained in practically their present form for use on the battleship Montana, now under construction at the Mare Island Navy Yard. Two additional brakes are to be fitted, making a

total of twelve. Although the probable launching weight of the new battleship is almost 50 per cent greater than that of the California, or about 20,000 tons, the necessary retarding effect is to be obtained by increasing the braking period, making use of the greater part of the channel fairway instead of attempting to stop the vessel in about 350 feet. Wire-connecting cables, secured to pad-eyes and lashed with small rope stops to the ship's side, will replace the former chain cables laid down on the slip, and the friction cables will be laid out in pipes or boxes instead of being wound on reels.

As a matter of fact, the unforeseen developments of the California launching were, in reality, the most valuable incidents that could have taken place. With the very high wind that was blowing across the fairway at the time of the launch it was indeed fortunate that the vessel stayed on the mud on the far side of the channel long enough for the fleet of tugs to make fast. Furthermore, the fact that the ship ran for about 250 feet with no other retarding effect than that of the water resistance has enabled the plotting of an accurate velocity-distance curve which would otherwise have been well-nigh hopeless of accomplishment. This curve, taken with the full-sized ship (cradle and all appendages in place), represents accurately what the author has tried to obtain experimentally with innumerable runs of the model.

In connection with the above, and referring again to last year's paper, Plate 13, it will be noted that the velocity of the model on Run No. 119, at 570 feet travel, was exactly the same as that of the California at the corresponding point when the brakes were applied, *i. e.*, 21.5 feet per second. A solution of the two formulae ($K_1 = VS$ and $K_2 = \frac{M}{VS}$) for corresponding points in the free run of the model and ship, gives the following results:—

	Model	*Ship*
Origin of curve from origin of velocity distance curve, feet	+27.2	—13.1
K_1	13,085	11,630
K_2	81.64	87.2

The constant K_1 represents the number of feet that the ship would run in free route before the velocity were reduced to 1 foot per second. The constant K_2 is that employed in the equation $R_w = K_2 V^3$ to obtain the approximate water resistance of the vessel at any given velocity. The constants obtained from the data of the California launch show that the ship had somewhat more resistance than the previous model runs would indicate, by about 6¾ per cent. The model cradle, however, in order to avoid undue complication and to permit easy and safe handling during the experiments, was not provided with the thousand and one minute appendages to be found on the larger vessel, such as wire rope frapping, spreaders, securing lines and wedges. Had the model been so fitted, it is considered that the water resistance would have been increased by fully 5 per cent, in which case the curves and constants obtained from ship and model runs would have been in remarkably close agreement.

A direct comparison of the brake-resistance curves of the model and the ship, and the plotting of the probable brake-resistance curve of the California is a somewhat difficult undertaking, in view of the fact that the average travel of the friction cables through the brakes was only about 137 feet. Any part of the travel during which a fixed number of the brakes were in operation was so short as to preclude an accurate determination of the effect of brake resistance on the velocity of the vessel. A reasonably accurate estimate has been arrived at in the following manner, with a second computation which is in close agreement.

Referring to Plate 10, upon which all the curves obtained from launching data are shown, it will be found that at 710 feet travel, when 78 feet clear of the ways, the ship was traveling at the rate of 17.1 feet per second. The deceleration at this point was 0.6 foot per second, indicating a total retarding force of 609,000 pounds. Of this the water resistance, computed by the formula $R_w = 87.2 V^3$, was 435,800 pounds, leaving the brake resistance as the remainder, 173,200 pounds. With two brakes in operation at 830 pounds pressure and four at 600 pounds pressure, this gives a coefficient of friction of about 0.129. As an extension and check of this computation, consider the behavior of the ship from 660 to 910 feet travel, at which point all the brakes had ceased to have further effect. It is found that of the total reduction in kinetic energy for this portion of the travel, 98,730,000 foot-pounds, the water resistance had absorbed by far the greater part, namely, 80,782,500 foot-pounds, leaving 17,947,500 foot-pounds as the energy absorbed by the brakes in action. Summing up the brake effect, in terms of hydraulic pressure in the cylinders and travel of the respective friction cables, it is found that the average coefficient of friction is about 0.123. This figure, which is but slightly different from that above, gives an average coefficient of friction of 0.126, as compared with the coefficient of 0.24 as found by the preliminary experiments of a year and a half ago.

It must be remembered, however, that these tests were carried out with the limited means then at hand, which permitted only a comparatively short pull on one friction wire at a rate not exceeding about one-half foot per second. It must be remembered also that the tests were conducted with only two cables, which presented clean, bright surfaces to the brake blocks on most of the runs. The friction cables in place at the time of the launch, however, were covered with a partly oxidized coating of heavy oil, just as they had been delivered by the contractor, and the presence of this oil coating may possibly have reduced the friction by an appreciable extent. It appears quite likely, therefore, that at the speeds and temperatures at which these brakes run during a launching operation there is a considerable reduction in the assumed coefficient of friction, although the data given above are unfortunately neither sufficiently accurate nor extensive to fix this value within very close limits.

In connection with the launching of the battleship Montana the model is now being rebuilt and the apparatus slightly modified. The tank will be increased in width from 40 inches to about 72 inches and the new model will be provided with all the minute appendages in addition to the launching cradle. The subject of

stern masks having been brought into prominence during the discussion on the previous paper, some experiments will be carried out on the Montana model with masks of different sizes and shapes. As an example of the effectiveness of any means of increasing the water resistance, it may be noted that for the California (hull and cradle alone) the value of this resistance at the moment that the ship left the ways was 667,000 pounds, or just short of the total normal brake pull on the ten brakes.

DISCUSSION.

THE CHAIRMAN:—Gentlemen, you have heard this paper, "Launching Ships in Restricted Waters," which I think you will agree has been interesting, and the motion pictures unusually so.

The question of stopping ships after launching is one of considerable interest. In times gone by, our ships have usually been launched in waters where there was ample room, without the necessity for elaborate precautions to stop them at a short distance from their berth; but with the increasing size of ships this matter has become more and more important, and I think the Society is to be congratulated on having such complete records presented to it which show the necessary precautions to be taken for the launching of these very large vessels, where the launching space is so restricted.

The paper is now open for discussion. Mr. Gatewood, who has favored us with some papers on this subject, will probably be able to give some interesting remarks at this time.

MR. WM. GATEWOOD, *Member* :—I take great interest in the problem of launching vessels, and although at Newport News we have unlimited water and do not have to restrain our vessels when they are launched, it seems to me that the real trouble with the launching of the California was due to the fact that the declivity of the ways was too great. I think that when the vessel was clear of the ways there was stored up in it an enormous amount of energy, and that it was unnecessary for the purpose of the launching to have put into the vessel all this energy. So, if the launching of the California is to be a lesson for the launching of other boats, I should say that the lesson would not lie in improving the brakes but in so arranging the declivity of the ways that there shall be just sufficient velocity to cause the vessel to float, but not sufficient to have stored in the vessel as much energy as was the case with this vessel.

THE CHAIRMAN:—Are there any other comments? The nature of the paper is not such as to lead to very much discussion.

MR. H. P. FREAR, *Member of Council:*—At the Union Plant of the Bethlehem Shipbuilding Corporation, San Francisco, Cal., not very far from Mare Island Navy Yard and in waters quite as restricted, many vessels have been launched. Prior to the dredging in front of the plant being taken over by the State Harbor Commission, the mud was deep and at low tide partly exposed. At that time it was the custom to depend on the mud for retarding vessels during launching. As the mud filled in rapidly after dredging

it was necessary to dig a channel in front of the building slip about the length of the vessel each time before a launching. According to conditions, the vessel before coming to rest would bury herself about half a length in the mud at the end of the channel and after lying over one tide was easily pulled out. This was a most harmless and effective method under ideal conditions and embodied all the elements essential to a proper solution of the problem.

Work must be performed to retard a vessel, and work is measured by the product of resistance into the distance through which its point of application is moved. Distance is equal to velocity into time, therefore time is an important factor in the retardation of a vessel. As the time during which a vessel is being retarded increases, the required resistance decreases. If a greater part of the distance available for retarding the California had been utilized as suggested by Lieut. Commander Saunders, less resistance would have been required, in which event the breaking of chains and cables might have been avoided. The snapping of chains or breaking of cables has little or no retarding effect on a vessel unless accompanied by substantial stretching so as to include an element of time during the breaking process. The method employed in retarding the California possessed in a high degree all the essentials necessary to accomplish the desired results. A similar method has been used for some time at the Bethlehem Harlan Plant, Wilmington, Del., but in lieu of wire cables 9-inch Manila hawsers are used. The frictional resistance or retarding force is effected by means of a wooden clamp.

The climatic conditions in California are perhaps more favorable to launching than in the east, where it may be either excessively hot or cold, and on that account there is less liability of a vessel sticking on the ways. At the Bethlehem San Francisco Plant, where the pressure per square foot of ways was about the same as in the case of the California, a mean declivity of about ½ inch per foot at the start was usually adopted. It seems as though a mean declivity of 11/16 inch per foot, as used in the case of the California, was excessive for such a heavy vessel, especially where the waters were restricted.

Mr. R. Walter Creuzbaur, *Associate:*—As a civil engineer, who had been quite unfamiliar with some of the methods of launching ships in restricted waters, when I was put in charge of the civil engineering work at the Hog Island Shipyard and Bristol and Newark three years ago, the matter of outshore dredging, to provide sufficient run for the ships, was important and pressing. The dredges throughout the country were largely occupied, and in certain sections the prices on dredging were exorbitant.

At Hog Island there were six and one-third million yards of dredging. It was laid out most generously and the question of the extent of the ship's run after launchimg did not come up as a serious question, at least to the agent. At the Newark Bay Shipyard, at that time, with Admiral Rousseau I cut out a million yards of dredging on the assumption that three times the water-line length of the hull would be sufficient for the run of the ship from the end of the way. That restriction was much criticised, particularly by the people who thought they controlled the dredging prices. We got the War Department dredges in there and materially cut the cost of dredging for the work actually necessary. We did not dig that million yards of mud and it was never necessary; that is the point I wish to make. We were ready for launchings, and, though the declivity of the ways was somewhat too steep, in no case did the ships run far enough to give any trouble. Without any retarding arrangements and a ¾-inch declivity three times the length of the hull was sufficient for the fairway.

We civil engineers of the shipyards at that time were suffering from lack of knowledge of just this sort of thing, which Commander Saunders deserves great credit for giving us now, completing, I understand, his first paper.

We tried to economize all we could at the shipyards in the matter of adequate launching room and cut down the dredging again at Bristol, Pa. We wanted to be safe, but did not dare to jeopardize these ships. We felt every one of them was a needed contribution to the success of the war, and we could not take a chance by trimming too much.

The launching arrangements were crude. On one occasion a number of distinguished men were present at a supposed launching at one of these shipyards, and the ship did not move—would not go down. I believe she had been down on the grease a little too long, but there was some doubt about the reason. On the other hand, I have seen them "burn up" the ways at the end of the run as they took the water. These two extremes show the crudity of the general arrangements; even a change of lubricant might have equalized the speed which was wanted.

This matter of being able to limit the run of the ship is important. It was extremely important three years ago with our one hundred and eighty shipyards for the merchant marine being built or extended on water fronts often restricted in width.

Mr. Edwin B. Sadtler, *Member* (Communicated):—The writer, while superintendent of the Harlan & Hollingsworth Yard at Wilmington, Del., in 1901, made use of weights dragged by the ship for checking same when launching the S. S. Denver, as follows:—

Weight of ship when launched, 2,650 long tons.

Two weights were used on each side of ship, one of 25 and one of 30 tons, or 110 tons in all, equal to 4 per cent of the ship's weight.

The weights were made up of punchings in rectangular boxes of plates bolted together and connected to ship by her own chains 2 ⅛-inch diameter of such a length that the 25-ton weight nearest the water dragged first, the 30-ton weight not starting until the former had moved some distance. As a shackle on the 25-ton port weight broke in starting, the 30-ton weight dragged over the stationary 25-ton port weight, the latter with the 25-ton starboard weight checking the ship before the starboard 30-ton weight started, or after the port weight had dragged 125 feet. Hence the ship was checked by a total weight of 55 tons or a little over 2 per cent of her weight.

Being associated with the Oscar Daniels Company, of Tampa, Florida, who are building 9,500 deadweight ships for the United States Shipping Board, I recommended drags on launching these ships, as two of their ways are restricted in launching space. We used the ship's chains for weights, coiled so as to drag the chain down a track over the ties, the chain being attached to the ship by wire hawsers.

The ships weighed 2,735 tons, and the chains (2 5/16 inches) arranged in two piles on each side of ship weighed 36 tons or 1.3 per cent of the weight of the ship. This weight dragged 300 feet before checking the ship. The next ship having a little less room to run, we increased the weight to about 1.6 per cent of the ship's weight, which checked the ship in 200 feet drag of chain. These chains were not lashed together in a pile, hence only 300 × 4, or 1200 feet of chain stretched out, were dragging on the first ship, and 200 × 4, or 800 feet, in the second, which would indicate a lower launching speed on the second ship as less weight of chain was being dragged. I regret no record of launching speed was kept. The launching ways had ⅝ inch incline per foot in each case.

The writer has had experience with brakes, dragging weights as above and slewing the ships by lines attached to the ship near the stern and led to a point up or down stream from the launching slip, but owing to the doubt as to the amount of friction created by the brake, prefers dragging weights wherever possible, except for very light ships.

Lieut. Commander H. E. Saunders, C. C., U. S. N. (Communicated):—Although the paper deals particularly with methods employed to check the speed of vessels after they have been launched and are water-borne, it is true that at any launching a consideration of this subject must also take into account the methods employed for sending the ship off the ways and into the water and the probable speed which will be attained in the course of this operation.

Considering the subject of launching a vessel as a whole, therefore, it must be admitted that the first and most important principle of a launch is to get the ship safely into the water, whereas the matter under discussion resolves itself into the most satisfactory, safe and economical method of stopping the vessel without damaging the ship or endangering life or property.

It may be said, therefore, in connection with the launching of the battleship California at Mare Island, that this vessel was almost three times as heavy as any which had been launched from the same ways, and that, with a ship so large and so costly, any sticking or stopping on the ways when partly water-borne was not even to be thought of.

As a matter of fact, the declivity, as is often the case, was determined partly by consideration of the limits for piling, headroom, etc., in the vicinity. Even though this declivity ($\frac{11}{16}$ inch per foot) may appear to be steep for so large a vessel, it is believed that the decision to adopt it was based upon thoroughly sound reasoning. It is certain that this argument was borne out by the facts, for the vessel was quite lively and started immediately upon release. At the same time there was no difficulty in holding her with the hydraulic triggers and no evidence of undue stress at any point, nor was there any doubt in the minds of those who were responsible for the launch that, once started, the vessel would take the water quickly and easily. What may have been thought rather a daring procedure was certainly the means of ensuring a most successful launch, whether the ship was afterwards fully stopped within the estimated distance or not.

Equally as daring may be considered the decision to retain exactly the same declivity for the battleship Montana, a ship which will be fully 50 per cent heavier when launched. However, the unit pressure will be reduced from 2.40 tons to less than 2 tons per square foot, with a considerable reduction to be expected in the launching speed. In this case, exactly as before, the matter of first importance is to get the ship off the ways and into the water; the question of checking her speed after that point must then be taken care of with the brakes and other means that may be at hand.

In the case of the Montana, the permissible travel after the vessel leaves the ways is almost exactly twice the length; it is considered that this should be sufficient distance for all vessels without undue expenditure of time and labor for brakes or drags, and a travel of three times the length should certainly be amply sufficient.

The Chairman:—In the absence of Commander Saunders, the thanks of the Society will be communicated to him for his paper.

As we have reached the hour for recess the Society will now take one. The afternoon meeting will begin at 2.15 with the discussion of the next paper.

SECOND SESSION.

THURSDAY AFTERNOON, NOVEMBER 11, 1920.

Mr. W. M. McFarland, Honorary Vice-President, called the meeting to order at 2.30 o'clock.

THE CHAIRMAN:—Before starting the afternoon session and having the next paper presented, there is a telegram from Admiral Capps, president of the Society, addressed to Mr. H. L. Ferguson as Acting President, which Mr. Ferguson asks me to read, as he cannot be here until a little later. The telegram is as follows:

"Cordial greetings to all. Best wishes for a successful meeting and great regret that I cannot be present.

"W. L. CAPPS."

This is from Seattle, where the admiral is now.

The next business on the program is paper No. 3, entitled "New 20,000-Ton Tankers," by Mr. Harold F. Norton, Member.

Mr. Norton presented the paper.

NEW 20,000-TON TANKERS.

By H. F. Norton, Esq., Member.

[Read at the twenty-eighth general meeting of the Society of Naval Architects and Marine Engineers, held in New York, November 11 and 12, 1920.]

The vessels designed by the Newport News Shipbuilding & Dry Dock Company for the Standard Oil Company of New Jersey will be of interest to the Society on account of being, so far as we are aware, the largest tankers yet built, and of several unique features in design, especially in connection with the structural work.

They are two twin-screw ships 550 feet between perpendiculars, 75 feet beam, 43 feet 3 inches depth to shelter deck, and about 20,300 tons deadweight on a draught of 30 feet. The vessels are designed with straight stem and elliptical stern and rigged with three pole masts. They are of the shelter-deck type with complete steel shelter and upper decks, except that the upper deck is omitted in the expansion trunk in way of the oil tanks. The vessels are being built under the special survey of the American Bureau of Shipping to the highest class.

The propelling machinery is located in the stern and consists of two triple-expansion engines set side by side in the same engine-room. The engines are designed for 3,800 indicated horse-power total at 200 pounds working boiler pressure and a piston speed of 585 feet per minute. The cylinders are 23-inch, 39-inch and 68-inch diameter respectively with a 45-inch stroke. The high-pressure and intermediate pressure valves are of the piston type, and the low-pressure valve is flat and double ported. The crank and thrust shafts are 13 inches in diameter, the line shafts 12½ inches and the propeller shafts 14½ inches.

One condenser is fitted outboard of each engine, with an attached air pump on each engine, and attached bilge and sanitary pumps on the starboard engine only. The starting platform is at the level of the top of the engine bed plates, and located between the two engines. There are two centrifugal circulating pumps with engines. The propellers are to turn outboard, one a left-hand and one a right-hand screw, each with cast-iron hub and three manganese bronze blades.

There are three single-ended Scotch boilers, placed side by side, with their backs toward the engines and fired from the forward end. They are four-furnace boilers 17 feet inside diameter by 12 feet long, and designed for 200 pounds steam pressure. So far as we are aware these are the largest Scotch boilers yet built in this country except those for three ships for the same owners built by the Harlan and Hollingsworth Co. in 1916, each of which had two boilers 18 feet in diameter by 11 feet 7 inches long, and which are said to have given excellent service. There is also a cylindrical two-furnace, single-ended, return-tube donkey boiler 10 feet

11 inches in diameter by 10 feet 10½ inches long. The main boilers are fitted for burning oil only, and the donkey boiler for coal only.

The fuel system consists of two horizontal duplex pumps 6 inches by 4 inches by 6 inches and two similar transfer pumps 14 inches by 9 inches by 12 inches. Three heaters are arranged so that oil may be discharged through them in multiple. The service pumps and their fittings are located in a gas-tight steel enclosure on the port side of the boiler-room. The transfer pumps are located, one in the fire-room and one in a special steel pump-room enclosure in the after end of the fore hold. The after pump has a 6-inch connection to the after fuel-oil tank and a 6-inch discharge connection to the 6-inch fore and aft transfer line to the forward fuel-oil tank and transfer pump, and also a 6-inch discharge connection on the shelter deck. The forward pump has a 6-inch suction to the forward fuel-oil tank and a 6-inch discharge to the transfer line and to the shelter deck.

There are two main feed pumps, 14 inches by 9 inches by 24 inches, with brass water ends and automatic control gear, each to draw from the main feed tanks and reserve feed tanks and discharge through the feed heater or by-pass to the main and auxiliary checks on main and donkey boilers. An auxiliary feed pump of the same size is fitted. There are also two injectors connected to the feed lines. There is an independent bilge pump, an engine-room ballast pump, a forward ballast pump, an independent sanitary pump, a donkey-boiler feed pump, a fresh-water pump and an evaporator feed pump.

Besides the above auxiliaries there are two evaporators capable of producing 35 tons of fresh water each 24 hours, and a distiller for 2,000 gallons a day. The main electric plant consists of two 20-kw generating sets in the engine-room, and there is also an emergency generating set of 12½-kw capacity driven by a gasoline-kerosene engine located on the upper deck forward. This set is also connected to the radio set, so as to provide for the lighting of the ship and sending of wireless messages even though the main engine-room may be flooded or on fire. The refrigerating plant consists of a 2-ton ammonia machine fitted in a gas-tight compartment in the engine-room.

There is an engineers' workshop adjacent to the engine-room equipped with a 14-inch screw cutting engine lathe about 6 feet between centers, a 20-inch drill press, and a 12-inch double emery grinder, all driven from a countershaft connected with a 5-horse-power motor.

The cargo oil pump-room is located amidships and fitted with four horizontal duplex main pumps with compound steam cylinders 12-inch and 18-inch diameters and 24-inch stroke, and oil cylinders 14½-inch diameter. These pumps are arranged to work with the ship's steam at 180 pounds or shore steam at 70 pounds, having a by-pass connection admitting live steam to the low-pressure cylinder when working with low pressure steam. There are also two auxiliary vertical duplex pumps, 12 inches by 8 inches by 12 inches. The main cargo oil-suction system consists of two lines of 14-inch pipes, one on each side of the center-line bulkhead, for draining the after tanks, and two similar lines leading forward for the forward tanks. Each main tank is connected to both main lines of piping by 10-inch

valves and pipes. Cross connections are fitted in the pump-room to enable any main pump to draw from any compartment on either side and discharge into any other compartment. There is also a 6-inch auxiliary suction line with a 4-inch suction branch fitted in each tank.

Most oil tankers are fitted with only two main cargo oil pumps. The above four large pumps are fitted on these vessels with the idea of being able to discharge the cargo with unusual rapidity. The 6-inch auxiliary suction line is fitted with the idea of being able to clean out the tanks after the large pumps have taken out the major portion of the oil. The size of the pumping equipment is evidence of the policy of the owners of obtaining the quickest possible turn around of the tankers in port.

There are four 12-inch discharge mains in the pump-room connecting to a 12-inch main leading aft on the upper deck, and to 12-inch cross mains on the shelter deck. There are also two 5-inch auxiliary discharges connecting to the by-pass of the main pumps. The discharge lines over the pump-room are cross connected so that any of the pumps can discharge separately to either or both sides of the vessel. There are two 12-inch sea chests with stop valves fitted to the ship's sides in the pump-room and connected to the pumps for ballasting the main cargo tanks.

Complete steam-heating system and heating coils for the main cargo tanks and fuel oil tanks are provided, and also steam-smothering pipes for fire extinguishing.

It will be noticed from the plans that there are eight main cargo oil tanks each 34 feet 6 inches long, and that the cargo oil pump-room is also the same length. This length of cargo oil tank is unusually great, but it will be noticed that the oil-tight center-line bulkhead divides each main tank into spaces on each side of the vessel which are approximately square; that is, the half beam of the ship, which is 37 feet 6 inches, is a little more than the length of a main cargo oil tank. There are no summer tanks, and the wing spaces outboard of the expansion trunk and between the upper and shelter decks are used for pipe passages only. This is due to the fact that the vessel is designed primarily to carry one kind of oil, and the omission of the summer tanks results in a simpler construction and a distribution of the structural material where it is more efficient in contributing to the longitudinal strength of the vessel.

The vessel is designed on the Gatewood system of longitudinal framing. The main framing consists of two deep transverse webs, ordinarily called transverses, fitted between each pair of main transverse bulkheads. The longitudinals are then worked continuously through these transverses and are cut and bracketed at the main transverse bulkheads. The Gatewood patented system of longitudinal framing takes advantage of the reinforcement of the longitudinals by the brackets at the bulkheads, and so spaces the transverses and designs the brackets as to make the maximum bending moments in the longitudinals as small as possible. At the same time some of the reaction load is taken from the transverses and transferred to the more rigid transverse bulkheads. Although the distance between

the main transverse bulkheads is 34 feet 6 inches, the transverses are spaced only 10 feet apart, thus making each transverse a distance of 12 feet 3 inches from the adjacent bulkhead. This dimension is 2 feet 3 inches greater than the distance between transverses, but results in no greater bending moments on the longitudinals than occur at the transverses, on account of the brackets which are fitted at the bulkheads. The other scantlings of the vessel in general are arranged about as usual for a 10-foot spacing between transverses and bulkheads, on a longitudinally framed vessel of the usual type.

The ordinary bottom longitudinals are 18-inch by 4.1-inch by 51.6-pound channels, spaced 33 inches, which is wider than on any previous longitudinally framed vessel, so far as we are aware. The side longitudinals gradually decrease in size to a 10-inch by 3.5-inch by 23.4-pound channel for the first longitudinal under the upper deck. The brackets connecting these longitudinals to the main transverse bulkheads are shown on the type plan of oil-tight transverse bulkheads, and the way the brackets gradually decrease in size to suit the size of longitudinals will be noted. The connections of the brackets to oil-tight bulkheads are all by means of 6½-inch by 6½-inch by 19.8-pound T-bars with the standing flange double riveted to the brackets, to provide the strongest possible connection and facilitate caulking.

The stiffening of the bulkheads is interesting on account of the unusual expanse of bulkhead to be supported. Deep bottom longitudinals are worked in way of the vertical webs *A* and *B* on the bulkhead, and web *A* is provided with a bracket and girder to the transverses at the level of the upper deck to take the bending moment at this point. The arrangement of these brackets and girders at the upper deck level to take the stresses transmitted from the webs of the transverse bulkheads is interesting, as will be noticed on the plans. There is a horizontal web at the upper deck level to take the reaction of the vertical web *A* at that point, and from the upper to the shelter deck in line with web *A* there is a small web designed to carry the load transmitted from the horizontal stiffeners in this location on the bulkhead. Web *C* is provided with a girder at the top and bottom to connect it with the transverses and take care of the bending moments in the brackets at the ends of the web. These moments on web *B* are cared for by its connection to the deep longitudinal at the bottom and to the expansion trunk bulkhead at the top.

Special transverse webs are provided in the wing spaces above the upper deck to provide support for the side and deck longitudinals in this vicinity. In order to provide as much longitudinal strength as possible, the side and deck longitudinals in this vicinity are worked continuously through all of the transverses, with butts staggered.

The rough and smooth sides of the bulkheads alternate, port and starboard in the main cargo tanks and on the center-line bulkhead through the main tanks, so that all the bulkheads may be tested and the smooth sides examined, after the ship is in service, by filling the minimum number of tanks. Both transverse bulkheads of the pump-room are arranged to be smooth on the inside of the pump-

room, so as to be readily examined at all times. In the same way all of the stiffening is kept inside of the fuel-oil tanks at the ends, which will normally have oil in them.

The sheer line of the vessel is straight and parallel to the base line for nearly the whole extent of the main cargo tanks. This fact, and the fact that the pump-room is the same length as the main cargo tanks, permit a rather unique arrangement of the shell plating amidships. Each midship shell plate from the keel to the stringer is the same length between centers of butts; namely, 34 feet 6 inches, the length of the main cargo tanks. All the butts are arranged symmetrically with reference to the transverse bulkheads and the transverses, so that a number of plates amidships on each strake are exactly like each other in every respect. An exception to the above is that the plates of the bilge strakes *F* and *G* are made shorter for convenience in rolling for the turn of the bilge, but even these are symmetrically arranged so as to be like each other in sets of three.

Another point of interest is that, instead of arranging the clips on the transverses symmetrical with the midship section of the vessel, as is usually done, they are arranged symmetrical with the adjacent transverse bulkheads in each case.

The straight sheer, and the fact that the pump-room is the same length as a main cargo tank, result in a particularly symmetrical arrangement of the center-line bulkhead plating and also of the deck plating. Another interesting point on the deck plan is the small size of the hatches and the small number of them, on account of the omission of the summer tanks.

The main fuel-oil tank is fitted at the after end of the main cargo oil tanks, and a cofferdam worked between it and the boiler-room. There is also a fuel-oil tank fitted at the forward end of the main cargo oil tanks. The space between the forward fuel-oil tank and the fore-peak bulkhead is arranged as a hold space divided into two compartments by a water-tight bulkhead to the upper deck level. There is a double bottom fitted under this space, arranged to be used for water ballast. A double bottom for feed water is also fitted under the engine and boiler spaces. The forward peak tank is arranged for water ballast. The after peak tank is arranged for fresh water.

Accommodations for the captain, deck officers, etc., are provided in a double tier of steel deck houses amidship above the bridge deck. A wheel and chart house of teak is built on top of the upper house and a navigating bridge fitted at this level. Quarters for the engineers and those for the petty officers are located on the upper deck abreast the engine casing. The seamen and firemen have ample and commodious quarters in the after end of the upper deck, including the rather unusual feature of a separate pantry, adjacent to the mess rooms.

A steel house is provided for the steering gear on the shelter deck aft and the top of this house fitted as a docking bridge. Four 26-foot metallic lifeboats are provided, all under mechanical davits. The vessels also have ample electric-lighting arrangements, cold-storage rooms, steam-heating system, interior-communication system, drainage system, fire-main and deck-service system, steam steering gear, steam windlass and winches. The plumbing arrangements are unusually complete for this type of vessel.

In fact the size, outfitting and equipment of these vessels are such as to make them a notable addition to the already very notable fleet which has been built up under the supervision of Mr. D. T. Warden, manager of the Foreign Shipping Department of the owners, the Standard Oil Company of New Jersey, and Mr. R. W. Morrell, their naval architect, who considers that these tankers, on account of their size and the simplified design made possible by adapting them to one trade only, will prove to be the most economical oil carriers yet produced.

DISCUSSION.

THE CHAIRMAN:—This paper, entitled "New 20,000-Ton Tankers," by Mr. Norton, is now open for discussion.

MR. H. P. FREAR, *Member of Council:*—The paper under discussion has been awaited by many of us with much interest, and our thanks are due Mr. Norton for his valuable contribution. Before referring to several points of the design, I think it will be in order if, as a representative of the Bethlehem Shipbuilding Corporation, I take issue with Mr. Norton regarding the statement that this is the largest tanker ever built. The combined ore and oil steamers Bethlehem has under construction have a real claim for this distinction, although the length is the same; namely, 550 feet. While the Newport News vessel is a little wider, the Bethlehem combined ore and oil carrier has greater depth and draught and a credit in her favor of about 200 tons deadweight when carrying either ore or oil in separate compartments. Now I am not claiming that we have a 41,000 deadweight tanker, but when we beat the Newport News Company on two counts the honors should go to Bethlehem. The Bethlehem ore and oil steamer was designed for a given deadweight while at one of the ports of call with an oil cargo, but her maximum assigned summer draught, which can be used at all other ports, will be somewhat greater, amounting to 28 inches more than Mr. Norton's vessel.

Mr. Norton also claims that the 17-foot boilers to be installed in his tanker are the largest built in this country, with the exception of two 18-foot boilers built at the Harlan Plant of the Bethlehem Shipbuilding Corporation. The Bethlehem Corporation is now building six 17-foot 6-inch boilers at one plant and nine 17-foot boilers at another plant, and this number may be materially increased.

The principle of taking advantage of the reinforcement of the brackets to reduce the space, and incidentally the size of the longitudinals, is well understood and has been quite universally used in Great Britain, since the advent of longitudinal vessels in the case of single transverse tanks where there is a bracket in every span.

A great deal has been said and done in the direction of fabricated ships and the duplication of parts. Mr. Norton directs particular attention to the fact that the pump room is located amidship, and both it and the plating are the same length as the cargo tanks; the inference, as I understand it, being that with these features, together with the straight shear amidship, there is a greater similarity of parts.

It would be interesting if Mr. Norton would investigate to see if the brackets which are increased to reinforce the longitudinals could not be made all of one size for duplication without sufficient increase in weight to outweigh the advantages. For instance, if the 15-inch longitudinals require a 15 rivet bracket, say for single riveting, or a 22 or 24 rivet bracket for double riveting, could sufficient weight be saved on the smaller longitudinals with span reduced by fitting the same over size bracket. If double-riveted brackets are adopted, they could all be punched the same, but as the bracket became excessively large for the smaller longitudinals, one row of rivets, or part of a row, could be omitted so that there might not be any greater number of rivets throughout. On the other hand, it appears evident that the smaller longitudinals require relatively larger brackets in order to equalize the bending moments.

What attracted my attention most in the design, however, was the longitudinal distribution of weight. The length taken up by the cargo, notwithstanding it is increased by the long pump room located amidships, seems relatively short, and the machinery space and void space forward correspondingly long. I would like to ask Mr. Norton if he found the great length of machinery space necessary for a suitable twin-screw installation or if, after arranging the distribution of cargo in the manner described, he then took what was left for the propelling machinery.

It is well understood that the greatest stresses in a tanker with the machinery aft are in the deck and in the sagging condition on account of the void space at the ends which, on that account, should be reduced to a minimum. While there has not been much trouble with longitudinally framed decks, there have been several catastrophies in the case of tankers with transverse framed decks. One of these broke in two in ballast with the ballast concentrated in three or four midship tanks, and another carrying molasses broke because all the space could not be used and the cargo was concentrated amidships. From this it may be concluded that the position of the load in a tanker may in some cases be more important than if increased but with better distribution.

It would seem that Mr. Norton might have claimed that the sagging moment was somewhat reduced on account of the large void space amidships due to the pump room, as this may be a redeeming feature.

Mr. Robert W. Morrell, *Member:*—I have very little to add to the very valuable paper which Mr. Norton has presented, and, in fact, he was good enough to discuss it with me before it was presented, so there are practically no ideas which I can bring out which would throw any additional light on the subject.

I might state, however, in connection with Mr. Frear's remarks regarding the distribution of the cargo space, that the cargo space was originally intended to provide cubic capacity for a quantity of heavy oil, and therefore made it necessary to distribute it over so great a length of the ship as would be necessary if the oil were of a greater volume. This naturally tends towards the shortening of the oil space and provides a greater space at the ends of the ship.

So far as the distribution of the weight is concerned, with relation to the sagging and bending moment, I am assured by the builders that the bending moments work out much more satisfactorily with this vessel than they have on vessels of previous designs. I think that is all I have to say on the subject.

Mr. T. M. Cornbrooks, *Member* (Communicated):—I was very much interested

to note in this paper that Mr. Norton refers to the Gatewood patented system of longitudinal framing, and cites as the feature of the same an odd spacing of transverses, making the wide spaces next to the bulkheads and fitting larger brackets to the bulkheads, so as to reduce the bending moment on the longitudinals.

I would like to say, as a matter of record, that this is not a new idea at all. While with the Bethlehem Shipbuilding Corporation (Sparrow's Point Plant), we constructed the first Isherwood ship built in this country. This was in 1909. It was a ship for A. H. Bull & Company, known as the Millinocket. Shortly after this we built two colliers, each 536 feet long, for the United States Navy, named the Orion and Jason. On these designs, which were furnished us by Mr. Isherwood, we used the odd spacing of transverses and the large brackets at bulkheads exactly as described in this paper, with the same idea in view. In fact, if my memory serves me right, this same feature was used in every ship built on the Isherwood system that we constructed at Sparrow's Point. The data for scantlings were furnished by Mr. Isherwood, so that it would appear that Mr. Isherwood antedated Mr. Gatewood by a number of years.

The Secretary:—Just at this point, I would like to ask Mr. Norton in his rejoinder to give me some information based on his practical experience, on a phase which I think is interesting. These tankers, as I understand, are built on the straight shear principle, practically throughout the whole cargo space, which is a large proportion of the ship's length. That has a distinct effect on the freeboard permitted by the classification society which means, on vessels of given dimensions and given hold depth, there is a certain lesser deadweight which the ship can carry, which may or may not be important in this particular case.

The arguments for this form of construction, as I understand them, are that it permits an actual uniform section of ship throughout the length of the parallel middle body, because there is no shear, and therefore all the transverses and bulkheads, as well as many shell plates and other items, are identical throughout this length; and it is claimed that there is a certain simplicity of construction secured in that manner, and therefore the accuracy of the work may be supposed to be greater than in a vessel of constantly varying depth. I wonder if the yards which have been building on that principle have really found that there is any material saving in cost in loft work, template work, and construction and building generally, following that method. I would like to know what the company thinks of it, irrespective of the demands of the particular owner for whom it may be building, whether or not that particular type of construction is or is not superior, whether from the owner's point of view or the builder's point of view.

Mr. Albert E. Saunders, *Member:*—I may help Mr. Cox a little on the subject of how much gain there is in the flat shear, etc. The first tankers which the Sun Company built were designed with 40 per cent of middle body with flat shear for the same extent. All the midship shell plates were kept the same length and width and large enough to take three longitudinals. They were so arranged that there were about 100 shell plates in each ship exactly the same and which could be used end for end or side for side.

The pump-room length was the same as the spacing of the transverses, so that the same could be placed to suit the requirements of the owners. One advantage was that it has been possible to adopt a design of vessel which is more or less a standard, and as owners demand different distribution of the bulkheads, the same are moved to suit the

spacing of the transverses and still retain the same arrangement as to plating, etc. The frames—that is, the transverses—were made of I-beams and the bulkhead bounding bars were of T's, and there again we had uniformity in shell connection. You can see in that particular case, where upwards of 100 shell plates were from the same template and where it was possible to punch and stack and use the plates as they came to hand, there must be a considerable saving. The flat shear penalized the ship about 6 inches on freeboard.

THE CHAIRMAN:—If there is no further discussion, we will ask Mr. Norton to close.

MR. NORTON:—Mr. Frear's remarks as to the size of ships and boilers and the other remarks remind me of the story of the old Scotchman who was found by a party of tourists sitting on one of the Scottish cliffs gazing out across the Atlantic. A young woman of the party spoke to him and said: "This is a wonderful view you have, away across the ocean." He replied, "Yes." She said, "I suppose on clear days you can see the shores of America." He replied, "Yes, you can see further than that; on a clear night you can see the moon." So it seems to me that many things depend entirely upon the point of view and the thing at which one is looking, and I shall not attempt to argue the question of whether our ships are the largest, or as to the priority of ideas in the design. The paper is intended only as a description of the vessels.

In connection with the matter of the cargo space, I may say that Mr. Frear is entirely correct, and this is a matter of compromise in the design. In these particular ships we tried to arrange things as we thought would produce the most desirable balance. We could have distributed the cargo over a greater length of the ship by increasing the wing passage spaces and lowering the upper deck, and in various ways, but we tried to make what seemed to us the most desirable balance of all of these features.

In connection with the matter of straight shear, I may say that we have rather committed ourselves to the straight shear on our recent tankers, thereby indicating that the preponderance of opinion in our yard is that the straight shear is desirable. I do not know that we can state exactly how much economy results, and there is some difference of opinion among us as to whether any great amount of economy really does result. But in a ship of this size we found that the penalty in freeboard with the straight shear was not sufficient to give us any trouble. The fact that all of the main oil-tight bulkheads through the central portion of the ship are exactly alike is of course of some advantage, and by making the pump room the same length as a cargo tank, which, as Mr. Frear remarked, helps the distribution of the cargo and at the same time gives greater symmetry to all of the construction, we believe that we have selected what is really a desirable arrangement of all these features.

There is one other little point that may be of interest. I do not know whether Mr. Frear has had the same experience, but we have had the greatest difficulty in persuading the owners, and the owners' representatives tell us they have had the greatest difficulty in persuading their captains, that if there are to be empty tanks in an oil tanker, the place to have them is in the middle of the ship. For some reason the captains cannot get away from the idea that the place to have empty tanks is at the ends of the ship. They seem to be used to seeing an empty space at each end of the ship. So we believe that if we had, right in the middle of the ship, right in front of their eyes, an empty space

exactly the same size as the ordinary cargo tank, perhaps they would get used to it and not object to having another empty space on either side of it.

THE CHAIRMAN:—If there is no further discussion, we will extend our thanks to Mr. Norton and will pass on to the next paper, No. 4, entitled "Economical Cargo Ships," second paper, by Mr. Alfred J. C. Robertson, Member.

Mr. Robertson presented the paper.

ECONOMICAL CARGO SHIPS—SOME MODEL EXPERIMENTS.

By Alfred J. C. Robertson, Esq., Member.

[Read at the twenty-eighth general meeting of the Society of Naval Architects and Marine Engineers, held in New York, November 11 and 12, 1920.]

This paper is in a sense a continuation of the paper of last year in which the writer analysed the effect of size and speed upon the economy of freight ships.

The purpose of this second paper is to present the results and particulars of some model experiments carried out with the object of arriving at the best form for propulsion of full cargo ships such as were referred to in the previous paper, and these results are placed before you in a form which it is hoped will readily admit of their application to the design of cargo ships.

The type of lines suitable for vessels of slow and moderate speeds and of full form has received much attention of late years at the various experimental model basins; and special mention is due to the very thorough series of tests made at the Froude National Tank, London, and reported through the Institution of Naval Architects by Messrs. G. S. Baker and J. L. Kent in several recent years, and to Mr. McEntee's recent papers presented to this Society, as well as to the earlier contributions to our knowledge of the subject made by Admiral Taylor, Dr. Sadler and other investigators.

In results of full form Admiral Taylor has shown that a certain amount of parallel middle body is necessary to produce minimum resistance, and Messrs. Baker and Kent have supplied the results of tank tests of vessels with 10 per cent, 30 per cent and 50 per cent parallel body located amidships.

Mr. McEntee's recent papers, confirmed by Mr. Semple (T. I. N. A., 1919), have shown that the location of the parallel body forward of amidships has diminished the resistance of the model and also increased the efficiency of propulsion.

This paper covers somewhat similar ground for vessels of several prismatic coefficients exceeding .70, covering the field somewhat more fully and also supplying some further information regarding the form of areas curve associated with minimum resistance.

The experiments about to be described were conducted in two series independent of one another. The first set, designated Model 1107, consisted in the modification of the curve of areas of the after body (both in its area and in its curvature) of a successful cargo ship of high prismatic coefficient. The second set, based on Model 1130, was more extensive and consisted in the development of several entrances and runs each of identical form as to cross-section and curve of areas but of varying length, and combined with various lengths of parallel middle body, and these various fore and after bodies so united as to secure two or

three models each of identical dimensions, draught and prismatic coefficient but with the position of the parallel middle body varied. These experiments were all carried out for the Emergency Fleet Corporation, and it is by their courtesy that they have been made available for publication to-day.

A tabulation of the particulars of both sets of models is presented herewith.

Particulars of Models.

Design 1107.

Displacement Length, 400 Feet; Breadth, 53.66 Feet; Draught, 23.85 Feet; Tested at Draught Ratio of .438

Type	Percentage parallel body	Prismatic coefficient	L. C. B. from amidships
			Per cent
Fore body	25.0	.837	42.47
After body A	23.7	.817	41.63
AX	15	.758	39.51
B	15	.790	40.51
BX	9.6	.758	39.33
C	15	.790	40.83
CX	9.6	.758	39.68

Design 1130.

Displacement Length, 400 Feet; Breadth, 52.70 Feet; Draught, 23.29 Feet; Tested at Draught Ratio of .435

Type	Percentage parallel body	Prismatic coefficient	L. C. B. from amidships
			Per cent
Fore body B	5.84	.709	37.8
C	10.95	.743	38.9
D	16.7	.776	40.1
E	21.19	.810	41.4
F	26.31	.844	42.8
G	31.42	.878	36.3
After body V	4.56	.690	37.2
W	9.5	.724	38.2
X	14.45	.758	39.4
Y	19.38	.791	40.6

The above percentages for position of L. C. B. are based on half length of ship, each type listed being one-half length of vessel. For convenience the parallel body is expressed as a percentage of the full length of the ship.

Method of Experiment.—These tank tests were, in almost every case, carried out at the University of Michigan experimental tank because models used in that tank are of wax, and variations in such models are much more readily accomplished and at a much lower cost than is possible with the larger wood models in Washington. At the same time, one single-screw model was tested at Washington in order to link up the Ann Arbor and Washington tests; and the twin-screw model, with and without bossing, was tested in Washington.

It should be noted that the method of extension of experimental model tank tests at Ann Arbor is based on the coefficients of Tideman both for the model and the full-sized ship. These apparently coincide with the results of tests made at the tank some time ago on the frictional resistance of plane surfaces. At the same time it must be pointed out that Tideman's formulæ for skin friction, when applied to the full-sized ship, give resistance results considerably higher than those attained by the use of the Froude coefficients in the region of speed of cargo ships such as are now under consideration.

Method of Presentation.—All the results are presented herewith in the form of C constant curves for a length of 400 feet, wherein—

$$C = \frac{427.1 \times \text{E.H.P.}}{V^3 \times \Delta^{\frac{2}{3}}}$$

and

$$\text{E. H. P.} = \frac{C \times V^3 \times \Delta^{\frac{2}{3}}}{427.1}.$$

This is somewhat of an innovation on this side of the Atlantic, but the advantages of this form of presentation appear to the writer to be so important that he considers it well worth while to draw the attention of the Society to this system. It should be noted, however, that the speed ratio $\frac{V}{\sqrt{L}}$ used in this paper is the one familiar to our designers, where V equals the speed in knots and L is equal to the actual displacement length of the ship. No use has been made of the speed constant P used largely by Messrs. Baker and Kent, because curves on that basis are not so readily comparable for estimating purposes and also because the writer has failed to detect any corroboration of Messrs. Baker and Kent's theory regarding humps and hollows of resistance in the models whose results are presented to you to-day. It is probable that the theory advanced, that at certain fixed P values resistances are at a maximum, may not hold good where the variation in prismatic coefficient is procured by variation in ratio of entrance to run in conjunction with variation of parallel middle body, or possibly the speeds of these ships were too low to prevent the masking of Baker's humps by other more important developments of resistance. I believe that the curves placed before you to-day would permit some investigation along these lines.

In order to illustrate this form of presentation and to permit of ready comparison of present results with previous investigations, diagrams of tests showing the effect of location of parallel middle body, as carried out by Mr. McEntee and

presented to this Society a couple of years ago, are shown on Fig. 1, Plate 15, and Fig. 2, Plate 16, gives Taylor's standard series plotted as C curves for 400 feet length, together with C correction curves for other lengths, and a curve of $\frac{427.1}{\Delta^{\frac{2}{3}}}$ designated X by means of which the effective horse-power may be immediately arrived at from one diagram. Areas of the various sections at different prismatic coefficients for Taylor's series are shown on Fig. 3, Plate 17, in order that the form of Taylor's models may more readily be compared with the series here presented.

In addition a diagram, Fig. 4, Plate 18, is presented, showing the theoretical difference between tank tests as extended by the various coefficients of friction now in use at Ann Arbor, Washington, and in England. These curves are based on the assumption that the actual friction of a model of unit length at the three tanks is identical, but careful comparisons between tests of the same model made at Ann Arbor and at Washington seem to indicate that the Washington resistances, particularly for models of high displacement, run somewhat higher than can be anticipated from these curves. Whether this is due to some interference in the stream lines, due to the cross-section of the Washington model being larger in proportion to the section of the tank than is the case at Ann Arbor, or whether it is due to elements of eddy making not yet properly understood, or simply to the actual difference in the surface of the models, is yet unknown to the writer.

Because of the uncertainty of what is the true frictional resistance of these models, the method adopted at Ann Arbor of using Tideman's coefficients has been accepted for use by the writer in extending present results, and we thus secure very conservative resistances.

That the Ann Arbor estimates of effective horse-power for slow-speed cargo steamers appear to be very close to the actual effective horse-power required for the ship has been demonstrated in quite a large number of cases, and I think this might be explained by Baker's theory of the augmentation of frictional resistance in full ships, due to the nature of the wave profile and the resultant acceleration of velocity of the water at the surface of the vessel, and the magnitude of this increase in resistance is roughly equal to the difference between effective horse-powers calculated by Froude's method and effective horse-powers calculated by the Tideman formulæ. I have not been able to ascertain whether Baker has applied this correction to his model resistances as recommended in his book* before compiling his C constants given in his recent papers.

All results given herewith have been standardized for a water temperature of 70° F.

Those who have prepared the lines of vessels with parallel body will recognize that it is difficult to state definitely the point where entrance and parallel body combine, and that this point may be varied very considerably with an extremely small variation in the form of the ship. For this reason it was found necessary, in order to make comparisons fair, that an analysis length of entrance and run should be devised. At the same time it has been demonstrated repeatedly that the form

*"Ship Form, Resistance, and Screw Propulsion," paragraph 10.

of midship section within the range adopted in cargo ships does not make any material difference in the resistance of the ship, and, as it was necessary to standardize draught as well as form, the following method was employed. A slight modification of the block model, generally known as Dr. Kirk's analysis, was adopted wherein the breadth of the model was maintained equal to the breadth of the ship, and modification was made in the draught, this being such that the cross-sectional area of the model is the same as the midship section of the ship. This gives us an analysis draught equal to the product of draught of the vessel and coefficient of midship sectional area, and the angles of entrance and run are such that the plan of the block model contains exactly the same area as a curve of sectional areas of the ship. The fraction of the length of the vessel which would consist of entrance is equal to 1 minus prismatic coefficient of fore body. The fraction of the length of the vessel represented by parallel middle body is equal to twice the prismatic coefficient of the whole ship minus 1, and the fraction of length of run is, of course, equal to 1 minus prismatic coefficient of after body. This block model, which is bounded by straight lines and contains a volume equal to the displacement of the ship, indicates the mean angle of entrance and mean angle of run of the vessel, as well as the analysis length of entrance and run; examples are shown on Figs. 5, 6, 7 and 9, Plates 19 to 22.

In order to distinguish this analysis entrance from the ordinary length of entrance, the Greek letter ϵ has been used and for run the letter ρ has been adopted, the parallel middle body being represented by π and the ratio of entrance to run has been expressed as ϵ/ρ throughout the paper. The analysis draught, when divided by breadth, gives the analysis draught ratio δ. (See Appendix I.)

SERIES 1107.

Model 1107 is that of an ordinary cargo ship with plumb stem and counter stern raised well above the load waterline, and the length for displacement was taken as 98 per cent of the length between perpendiculars. The original model designated *A* had a curve of areas as shown in the lower part of Figs. 5 and 6, Plates 19 and 20, and was tested at Washington and Ann Arbor. The Ann Arbor model was then fined in two steps as shown, the resulting resistance curves being also shown. Cross curves of C values at different speeds are given in the left-hand end of each diagram plotted on ϵ/ρ ratios, and a zero line, which required a certain amount of fairing, has been shown indicating the actual difference in C value for model A as tested and extended at Washington and at Ann Arbor. In Fig. 6 the two modifications designated C and CX are identical with B and BX in Fig. 5 as regards prismatic coefficient, but are straight in form and consequently easier at the after shoulder than the B curves. It will be noticed that the straight form generally produces the lower resistances but has not eliminated a curious crossing in the C curves around the region of a speed ratio of 0.50.

A third comparison of resistances is shown in Fig. 7, Plate 21, BX and CX being plotted together with an intermediate curve identical in form with the

original curve of areas A, but of the same prismatic coefficient as BX and CX. Special attention must be drawn to the development of the hump in these three models, all of identical prismatic coefficients, and the emphatic increase in resistance which AX shows compared with BX and CX at a speed corresponding to 10 knots for a 400-foot ship cannot be lost sight of, because this illustrates a danger to which naval architects are exposed in departing from known models from which resistances are estimated.

SERIES 1130.

Series 1130 represents a model of a single-screw cargo ship as shown on Fig. 8, Plate 22, which gives cross-sections of the entrance and run, the parallel body being omitted. This model has a very small cruiser stern which, at the draught reported herewith, was only sufficient to carry the lines and curve of areas to the aft perpendicular. Generally the aperture in a single-screw steamer makes the displacement length 2 per cent or 3 per cent less than the length between perpendiculars. In this series the various combinations of entrance and run produced in all 18 models for the single-screw type, and there was a twin-screw model of "DW" modified, both with and without bossing. The twin-screw model was tested at Washington only, and the "DW" was tested both at Washington and Ann Arbor. All these models were tested at draught ratios of 0.489, 0.435, 0.351, 0.261, with the exception of the twin-screw model, where the displacements were held identical and the draught ratio was therefore very slightly reduced; but in order to keep this paper within reasonable compass the results at $\delta = 0.435$ only are presented herewith.

Fig. 9, Plate 23, indicates the angles of entrance and run corresponding to these models, and standard curves of areas of the two extremes of the W series are shown. These standard curves of areas are plotted in such a way that the actual area of curve in each case is identical, thus indicating in what part of the ship the displacement is carried. For this purpose, the midship area is laid out equal to 100 divided by three times the prismatic coefficient. In Figs. 10, 11, 12 and 13, Plate 23, these resistance curves are presented where each after body is shown with three or four different fore bodies, one or two curves having been left out of some of the diagrams to prevent confusion. Resistance curves could equally well be assembled with identical fore body and varying after body, and the student will find advantage in laying them out this way.

In Figs. 15 to 20, Plate 23, the curves of resistance shown in Figs. 8 to 11 are assembled in accordance with the prismatic coefficient, and these curves will probably prove of more use to anyone desiring to utilize this series of experiments for design purposes. Cross curves of each of these diagrams can also be plotted similar to those shown in Figs. 4 and 5 and, if this is done, it becomes a simple matter to select the best ϵ/ρ ratio for any particular speed.

Fig. 14, Plate 23, shows the comparison between single-screw vessels and a ship of similar lines to DW with the aperture filled in as indicated by the dotted

lines on Fig. 8, and the resistance of the twin-screw bossing for this model is also given as indicating the augmentation of resistance which is produced on twin-screw ships.

In order to fully utilize these results in the design of a cargo ship, Fig. 21, Plate 24, was prepared. This diagram is laid off on prismatic coefficients, both for fore body and after body, which are independent of one another, and when the necessary prismatic coefficient of each end is chosen from one of the figures (15 to 20), it is immediately possible to lay off a curve of areas for the prismatic coefficient of each end from Fig. 21. The amount of parallel middle body is also shown so that the entrance and run may be divided up equally and the sections made to correspond to Fig. 8. As it is necessary, usually, in designing a ship, to have pretty accurate knowledge of the height of transverse metacenter at an early stage of the work, a new diagram, Fig. 22, Plate 25, has been devised by the writer. It gives the location of the transverse metacenter from the load waterline as a coefficient of the beam of the ship at various draught ratios and at three prismatic coefficients, but it is noteworthy that the prismatic coefficient has an extremely small influence on the stability at draught ratios such as are adopted in ordinary merchant ships. For example, a ship of, say, 50-foot beam at a draught of 19 feet 6 inches shows no change in metacentric height between the prismatic coefficients of 0.68 and 0.84, the metacenter being about 16½ inches above the load waterline in each case. This diagram is strictly true, of course, only for ships of form given in Figs. 8 and 21, but it will be found that it gives reasonably accurate results for any ship within the range of coefficients covered.

APPENDIX I.

LIST OF SYMBOLS EMPLOYED IN PAPER.

C_m = Midship area coefficient, *i. e.*, $\frac{\text{Mid area}}{\text{Breadth} \times \text{draught}}$.

C_p = Prismatic coefficient, *i. e.*, $\frac{\text{Displacement in cubic feet}}{\text{Length} \times \text{mid area}}$.

C = Froude's resistance constant = $\frac{\text{E.H.P.} \times 427.1}{V^3 \times \Delta^{\frac{2}{3}}}$.

C_f = Frictional resistance constant = $\frac{\text{F.H.P.} \times 427.1}{V^3 \times \Delta^{\frac{2}{3}}}$.

S = Wetted surface constant = $\frac{\text{WS} \times .09346}{\Delta^{\frac{2}{3}}}$.

W.S. = Wetted surface in square feet $\left(\frac{S}{.09346}\right)$.

V = Speed in knots.

$S\Delta$ = Standard displacement = $\frac{\text{Displacement tons}}{\left(\frac{L}{100}\right)^3}$.

L.C.B. = Longitudinal center of buoyancy.

E.H.P. = Effective horse-power $\left(\frac{CV^3\,\Delta^{\frac{2}{3}}}{427.1}\right)$.

Δ = Displacement in tons of 35 cubic feet (salt water).

δ = Draught ratio, *i. e.*, $\frac{\text{Mid area}}{\text{Beam}^2} = \frac{\text{Draught} \times C_m}{\text{Beam}}$.

ϵ = Analysis length of entrance = 1 minus prismatic coefficient of fore body.

ρ = Analysis length of run = 1 minus prismatic coefficient of after body.

π = Analysis length of parallel body = twice prismatic coefficient of ship − 1.

APPENDIX II.

Two examples of the use of this paper are herewith given in detail:—

(*a*) Find the E. H. P. for the following ship at 11.0 knots. Length *BP* and waterline, 420.0 feet. Breadth, 53.0 feet. Draught, 26.5 feet. Mid area coefficient, .987. Displacement, 13,000 tons. $\frac{\epsilon}{\rho}=0.768$.

$$\text{Block coefficient}=\frac{13{,}000\times 35}{420\times 53\times 26.5}=.7713.$$

$$\text{Prismatic coefficient}=C_p=\frac{.771}{.987}=.781.$$

$$\delta=\frac{26.5\times .987}{53}=.4935.$$

$$\frac{V}{\sqrt{L}}=\frac{11}{\sqrt{420}}=.537.$$

From Fig. 19 C for $EX=.720$; for $C_p=.784$. $\frac{\epsilon}{\rho}=.755$.

18 C for $EW=.691$; for $C_p=.767$. $\frac{\epsilon}{\rho}=.688$

Therefore, by interpolation for proposed ship with $\delta=.435$ $C=.715$

and with $\delta=.493$ $C=.704$

Length correction $-.003$

C for Ann Arbor .701

C for Washington .651

$$\text{And E. H. P. (Washington)}=\frac{.651\times 11^3\times 13{,}000^{\frac{2}{3}}}{427.1}=1{,}120.9.$$

The actual H. P. of above ship as tested at Washington was 1,122.

(*b*) What are the best proportions, etc., of a ship of 550 feet length and 21,720 tons displacement for a speed of 16 knots, and estimate E. H. P.

$\frac{V}{\sqrt{L}}=\frac{16}{\sqrt{550}}=.682$ and by inspection .767 C_p is the highest C_p possible.

$$\text{Then mid area}=\frac{21720\times 35}{550\times .767}=1802.$$

For $\delta=.435B=64.4$ feet and draught with $C_m=.98=28.58$.

From cross curves of Fig. 18 $\frac{\epsilon}{\rho}$ should $=.70$ and $C=.730$.

$$\epsilon+\rho=1.00-\pi=1.00-.534=.466.$$

$$\epsilon=\frac{.70}{1.0+.70}\times .466=.1919.$$

$$p = \frac{1.0}{1.7} \times .466 = .2741.$$

Therefore fore body $C_p = 1.0 - .192 = .808$. After body $C_p = .726$.

Thus the vessel should be 550 feet × 64 feet 5 inches × 28 feet 7 inches draught

C_p of ship $= .767$. C_p of fore body $= .808$. C_p of after body $= .726$.
C_m $= .980$. L. C. B. $= .463 = 254.7$ feet aft of F. P.

Height of transverse metacenter $= .023B$ below $WL = 27.1$ feet above base.

$$\text{E. H. P.} = \frac{CV^3\Delta^{\frac{2}{3}}}{427.1}. \quad C = .730 - .018 \text{ (Length correction, Fig. 2, Plate 16).}$$

$$\text{E. H. P.} = \frac{.712 \times 4096}{.546} = 5341.$$

This power would be reduced by increase of draught until $\delta = .485$ at which E. H. P. $= 5{,}220$ but there would also be a reduction in KM and stability.

If it is necessary to locate the L. C. B. differently it is possible from the diagrams to estimate the increase in H. P. due to this. If L. C. B. were moved 15 feet farther aft the power would be increased by 200.

DISCUSSION.

THE CHAIRMAN:—Paper No. 4, "Economical Cargo Ships," is open for discussion. This paper represents a great deal of work, and I am sure you feel indebted to Mr. Robertson for the pains he has taken to supplement his previous valuable paper by this one. I hope there are some members here prepared to discuss the paper and to give some added light on this subject. Perhaps Dr. Sadler will discuss the paper.

PROF. HERBERT C. SADLER, *Member of Council*:—I would like to draw attention to one remark on page 44 of the paper, which is as follows: "These experiments were all carried out for the Emergency Fleet Corporation, and it is by their courtesy that they have been made available for publication today." In these days of criticism of everything connected with the Shipping Board, perhaps it is rather interesting to note that after all, that branch of it known as the Emergency Fleet Corporation did do something in the way of constructive experimental work, which we hope will be for the benefit of the shipbuilders of the country. The results of these experiments are made public in this paper by Mr. Robertson today. I may say that there is another series of experiments which we have just completed, which will also be published at a later date. The results are well worthy of careful study.

I remember some years ago we tested two models, one after the other, in the tank at Ann Arbor. I was rather astonished to find that the one with the smaller prismatic coefficient drove harder, a good deal harder, than the one with the large prismatic coefficient. It seemed not quite right. In those days we had not carried out many

investigations on the distribution of displacement fore and aft, and this paper shows it does not necessarily follow that, by fining up a vessel, you can get one which will be more easily driven; in fact, if you examine the curves, you will find it is possible to add considerably to the displacement of a vessel, put the displacement in the right place, and get a vessel which will be more easily driven than one with less displacement. From an economical point of view, therefore, these results are very valuable.

With regard to the differences between the tanks, at first it would appear as if there was a disagreement. We have tested at Ann Arbor a model, actually the same model pulled in the tank at Washington—it happened to be a model of a submarine—and the resistance curve checked almost absolutely, showing, so far as the results are concerned, the tanks are in agreement.

The difference enters in the allowance for surface friction, and that matter, as you know, has been discussed by Mr. Baker and by those on the other side, and is still perhaps somewhat open to doubt in comparing the model results with the full-sized ship. My own experience at Ann Arbor has been, in the results we have obtained and in checking them over with some vessels in actual service over a long period of time, that we get results which agree very well with practice. The Tideman's coefficient of friction which we use seems to hit the case pretty closely.

I may refer to one other fact which has been also the subject of a certain amount of controversy, and that is the shape of the sections of a ship, forward and aft. There seem still to be certain people who favor the club-footed section with straight sides, as against the section of the more V-shaped rounded form and flared-out side. I still have my own opinions in the matter as to which is the best form, particularly for full vessels; and in order to settle the matter we are conducting a series of experiments which, taken in conjunction with those which have been presented to us in the paper, will probably settle the question as to the most economical form of slow-going vessels for the future; that is, we are varying the shape of sections, keeping everything else constant.

I would like to raise one point in the meeting today, and that is in connection with the method of presenting results. Mr. Robertson has chosen the Froude method, the C-constant value, and that has been open to a great deal of criticism, particularly from shipbuilders and naval architects who have to use it. If you are used to that method it is a comparatively simple thing; if you are not used to it, it causes a good deal of trouble, and I would like to have an expression of opinion from the members of this Society as to what they consider the most desirable way of presenting results. One of the main purposes of a paper like this is to get the results before the shipbuilding fraternity and have them in such a form that the people interested will feel they can use them readily. If there are any suggestions to be made along that line, I think Mr. Robertson and I will be glad to hear them.

Admiral Taylor and I adopted in the early days, if you will remember, the method of presenting results in the form of pounds per ton of displacement, and that seemed to us a rather simple method. The method of Froude, which Mr. Robertson has adopted, has certain advantages when you are used to it, but the thing is to get used to it.

Mr. E. H. Rigg, *Member:*—This paper by Mr. Robertson is, like his paper of last year, extremely interesting. Those of us who have been associated with the design of slow-speed ships for some years must have been recently struck by the fact that the field which was practically devoid of definite and valuable data for guidance, is now beginning

to be well stocked. I say practically devoid, advisedly, because there always was some information available. The field of fine models, high-speed vessels, war ships and passenger liners has been well explored before cargo models were taken up seriously and consistently. We must thank investigators like Dr. Sadler, Admiral Taylor, Mr. Robertson, also Messrs. Froude, Baker and Semple on the other side, for the large amount of model-tank information which is now available in the field of slow-speed cargo-ship models.

Cargo boats used to be dismissed with a wave of the hand as being more or less beneath the notice of such austere institutions as experiment model basins, but the commercial world can congratulate itself that such is no longer the case. Mr. Robertson's present paper is another valuable contribution to our knowledge.

With reference to Professor Sadler's remark that the full shape is not necessarily the hardest driven, we have a case in point from experience at the New York shipyards some five or six years ago; we thought we had a bow that was too full—that is, some of us thought it was too full, but others thought it was all right. We tried the full bow and got certain results; we then cut it down to meet the ideas of the critics and tried it minus 200 tons of displacement. We got identically the same results up to some three knots faster than the ship was to be engined for; it is scarcely necessary to say that we built to the fuller bow and obtained the extra 150 tons or so of deadweight. The only thing the tank did not show up was her behavior at sea, as affected by the fuller bow. But we have had no complaints, in fact, nothing but praise, as to the seagoing qualities of that ship, besides several orders for duplicate vessels.

Mr. Robertson's figures throw light on the different methods of calculating results at the National Physical Laboratory (England), the Washington tank, and the tank at Ann Arbor, Michigan. Those of us who used figures from these several tanks have known for some time that the results did not check up well when comparing figures for slow-speed ships. This was sometimes very noticeable on checking a set of figures obtained from Washington data with the same ship figured by Ann Arbor methods. I think we can congratulate ourselves that this source of uncertainty is about to be removed. If you have a correcting factor, such as Mr. Robertson gives, which enables you to interpret the results worked by the method in vogue at the different tanks, you do not mind where you get your figures, because you can bring them to a constant basis and make them comparable.

Professor Sadler asked for some comment on the Froude constant method as compared with the pounds per ton. I think Mr. Robertson has made the constant method considerably more acceptable to those engaged in the practice of naval architecture by eliminating what are generally called the circle P values—that is to say, he plots his results on a base of speed-length ratio. If you eliminate circle P, you eliminate half of the terrors of the constant system. Otherwise I think the direct method of the constant, giving the whole resistance at once, is preferable for everyday work. For study of model forms, comparing one with the other, it will always be necessary to keep skin and residuary resistances separate. For routine work the value of the whole resistance at one step is acceptable; but there will be many cases, such as passenger liners, also high-speed bay and sound boats, where the skin and residuary resistances must be studied separately.

There is no doubt that this paper adds appreciably to the knowledge now available to the designer of full form vessels.

PROF. EDWARD M. BRAGG, *Member*:—A paper like this requires a good deal of study to get the meat out of it, but I have had a chance to look into the matter quite closely, and I think perhaps a few general conclusions may be of interest to this meeting.

One of the outstanding results of this investigation, to my mind, is this—it shows that for the speed at which cargo ships are ordinarily operated, say from .5 to .6 speed-length ratio, that the length of the run should never be less than 42 per cent of the length of the ship, and a very favorable length of entrance is 28 per cent. If you use an entrance of less than 25 per cent, the resistance increases very rapidly. This favorable condition is shown in Fig. 11, Plate 23, by the curve marked EW, which was obtained from a model with a parallel middle body of about 31 per cent. The run was about 42 or 43 per cent. and the entrance some 26 or 27 per cent. You will notice that it is the best curve over quite a considerable range of speed-length ratio from .55 to .65.

One rather curious fact is shown by the other curves. You notice that DW crops down below EW at the lower speeds. DW was obtained from a model which had about 25 per cent parallel middle body, and a longer entrance than EW. That curious fact seems to hold, that for the rather low speeds you want a fairly long entrance, and as the speed increases to speed-length ratios of .5 or .6, the entrance grows shorter, and above that value the entrance gets longer again, for the best results.

Of course it may be said it is not always possible to get in a run of 42 per cent; that it calls for such a fine run, particularly if you have the machinery aft; that you will not be able to get breadth enough for the engine bed. That may be so; but I wonder if it is always taken into account, if a certain type of machinery calls for a full after body, that machinery must be capable of developing 10 per cent more power; and should not that fact be considered in striking a balance between the advantages and disadvantages of that arrangement?

Take two vessels of 30 per cent parallel middle body, and in one place your parallel middle body at midships, and in the other shift it to the position I have indicated. There is easily a difference of 10 per cent in the power required. The one with the parallel middle body symmetrical with midships will drive 10 per cent harder than the ship with the longer run, and any type of machinery which calls for the full after body must have charged up against it the fact that this after body will require 10 or 12 per cent more power than the easier after body.

Many more points may be brought out here. In Fig. 9, Plate 23, we have the results given in terms of ε and ρ, and you will notice in almost every case the curves marked with values in the neighborhood of 1 drive harder. I think nearly 50 per cent of the models that come to the tank at Michigan to be tested have the entrance and the run of about the same length. The fact that the run should be longer than the entrance does not seem to have penetrated into all the shipyards and to have reached those people who have the designing of the lines in their charge, and I am wondering to what extent the results which are presented here get into the shipyards anyway. What measures do those in charge of the shipyards take to see that such results as these are available to their draughtsmen and designers? In how many shipyards can you find the complete transactions of this Society, to say nothing of the foreign societies which have contributed to this question for the last fifteen years? Shipyards are willing to spend thousands of dollars on recreational features for organized labor, but when it comes to spending five hundred dollars for the good of the draughtsmen, that is different. Five hundred dollars, I think, would buy all the transactions that are necessary, and I believe that would be

one of the best investments a shipyard could make. These transactions of the principal societies, placed in the drawing offices where the draughtsmen could get at them, would return 100 per cent profit every year.

There is another point I want to make—you men who are connected with the administration of the yards should see to it, for your own benefit, that these results penetrate to the men in charge. We know that in many cases the design of the lines is turned over to a man who has some whimsical method of design, and it is not uncommon for him to spend a great deal of time figuring whether he will put a waterline one-thousandth of an inch on this side, going through a point on this station, or one-thousandth of an inch on that side going through a point on the next station, when we all know the actual line on the ship as built will not come within an inch either way. It would be better to spend fifteen minutes getting the proper longitudinal distribution of displacement and then draw in the waterlines freehand than to start with a set of lines initially bad in this regard and try to improve them by local fairing.

Prof. C. H. Peabody, *Vice-President:*—I am speaking on this paper because it refers to Tideman's coefficients, which are frequently used with profit. I have no objection to urge against them if anyone finds it convenient, but I find works on naval architecture frequently refer to his coefficients as though they were obtained by him originally. I myself supposed that to be the case until one time a Dutch engineer visited our place and most kindly sent to us Tideman's work, which is a very large and important volume in Dutch. Fortunately, we had a well-informed Dutchman at the institute, who, though not familiar with naval architecture, could at any rate read this work to me, and in it it said positively he did not make any tests. There are many people who think his results were original, but he states positively in that book that his results are derived from Froude's work. I have many times wondered at Froude's courage in undertaking his work on friction and also why other people have not a like courage and give us more modern values.

Prof. Lawrence B. Chapman, *Member:*—I think the writer of this paper is to be congratulated, as Professor Peabody has congratulated him, on the excellent material he has given to the profession of naval architecture. It is rather unfortunate that powering data are presented in so many different ways. First we have Taylor's method of plotting, which is residuary resistance in pounds per ton against $\frac{V}{\sqrt{L}}$ and then Baker's method of circular C against circular P, and then Semple comes out with a circular C against circular L, and now Mr. Robertson's paper comes with circular C against $\frac{V}{\sqrt{L}}$ As Mr. Rigg says, Mr. Robertson has taken a good deal of the terror out of the circular P by taking the prismatic coefficient out of the radical, and it is a big advance.

It is easy enough to use all these methods, and at Lehigh we make the students use them all. They at least know how to handle them when they get through. It seems to me, from what experience I have had, that the Taylor method of plotting resistance on speed lengths against pounds per ton is the easiest method. You do not have to get the horse-power of the ship to make a comparison with other models, but in the other methods you must work up the horse-power before you can make a comparsion.

In the Taylor method and also in the method of Dr. Sadler you can get the frictional horse-power out of the way and begin to make a study of the residuary per ton for the various coefficients, and finally, at the proper coefficient, get the desired residuary per ton and thus do not have to work out the effective horse-power for each model.

It seems to me that the Taylor and Sadler methods of working with pounds per ton are more satisfactory, can be extended either to a large boat or a small boat, and are theoretically correct. The method of using the circular P works nicely for a boat of a certain size and within certain changes of lengths.

THE CHAIRMAN:—Is there any further discussion? If not, we will ask Mr. Robertson to close the discussion on the paper.

MR. ROBERTSON:—Professor Sadler drew attention to the diagram, also referred to by Mr. Rigg, on Fig. 11, Plate 23, which I think is perhaps the most important diagram presented here. It is an attempt to explain the difference existing in the tanks between this country and abroad. Mr. Rigg tells us that these differences have been known for some time. It has been extremely difficult to investigate that point thoroughly because the quantities with which we are dealing really come to the frictional resistance of the model, and that model resistance may total about one-tenth of a pound. The discovery of a 5 per cent difference in the friction of that one-tenth of a pound needed a good deal of careful comparison and investigation, and I just wish to say that the curves, though theoretically correct, are not actually correct. The models tested at Washington and at Ann Arbor did not test just as shown on these curves. In the case of the Washington models of light draught, small cross-sectional area, the difference is approximately what is shown here; but in the load draught portion, where the cross-section of the Washington model is larger in proportion to the cross-section of the tank than at Ann Arbor, the Washington model seems to test high, and, of course, that is a thing which will have to be investigated further—we are just feeling our way to eliminate some of the causes of the distrust of tank tests.

Mr. Rigg referred to the elimination of the P value in presenting the results by the Froude method. Of course the P value is not a Froude method. The P value is a Baker-Kent method, and I think that the very best proof of the value of that P can be obtained by the analysis of the results given in this paper. I mean to say that you can substantiate the value of the P, or discover it is not of as high value as at first we thought, by a careful study of these tests, where the parallel middle body was varied through so many stages as it was in this series of experiments. My own opinion is that in these tests, where the entrance and run proportions are not the same, the P value does not hold good.

I am very much obliged to Professor Peabody for throwing a little additional light on the Tideman's coefficient. Most of us who have been using them for a time were aware that Tideman had not actually carried out an intensive investigation, such as Froude had done, but he did give the world coefficients at an early date, which were adopted as if he had, and I think Mr. Froude deserves more credit than he gets for the frictional coefficients, even if in this country Tideman's slightly higher coefficients have been adopted, both at Washington and at Ann Arbor, for the full-sized ships.

Professor Chapman complains about the form in which the results have been presented, the circular C values. Of course the only difference between my form and those of Dr. Sadler and Admiral Taylor is simply in the C, because we use the same abscissa

values of speed-length ratio. Professor Chapman will see, when he investigates further, that there is a very distinct advantage, that he cannot, as he thinks he can, deal with the frictional horse-power and then have it out of the way altogether. If he takes a model of 75 per cent prismatic coefficient, he will find that by varying the position of the parallel middle body he varies the wetted surfaces, and, therefore, after choosing a good parallel middle body position and working out his residual resistance on that, he gets back and finds he has so varied the wetted surface, it is not the best. With the use of C values you can see at a glance exactly the best value for the whole ship, and, what is more, get at the results in much less time, and hence the C values have been presented to you.

Of course Dr. Sadler and others of us know that it is not the simplest thing to plot the C values from pounds resistance curves. There are certain corrections to be made, dependent on the size of the model and size of the full-sized ship. Once that is done, the work is practically all done, because the work of correction in passing from 100 to a 400 or 800-foot ship is very small and can be laid out as a simple curve as is plotted at the foot of Fig. 2, Plate 16, and the curve X also shown on that plate allows you at once, with the formula on the edge, to arrive at the exact effective horse-power for the model you are dealing with.

Mr. G. S. Baker, *Superintendent, Teddington Tank, England* (Communicated):—There are three points which arise in the paper which touch all experimental tanks. First we owe our thanks to Mr. Robertson for the data which he has given which considerably increases our knowledge in certain directions and will undoubtedly be useful for design work. Secondly, with regard to the correction of skin friction in passing from model to ship. I am a little lost as to what is actually happening in America in connection with this important matter. So far as I am aware there is in existence only one genuine set of experiments on the friction of planks in water. There were made by W. Froude. They were corrected by his son and amplified by experiments made by Gaber and at the Teddington and Washington tanks. These experimental results are used at the Teddington tank for all skin-friction correction work.

No experiments on skin friction were ever made by Tideman, and the statement which appears on page 46 of the paper that the method of correcting is by the use of Tideman's coefficients can only mean that the coefficients used are those adopted by Ann Arbor taken from those given by Tideman, which purport to represent Froude's results in another language. It would seem so much more preferable to use, as we do at Teddington, the original experiments of Froude for one's correction data than to use this longer and necessarily less accurate method.

With regard to the comparison between tests of the same form made in different tanks, we were able through Mr. Luke to test a model at Teddington which had already been tried at Washington, and some years later through Admiral Taylor a second form was tested in the two tanks. In both cases, when corrections had been made for temperature and length differences, the results of the two tanks agreed very closely over all the useful speed range. Comparison was here made between the ordinary Washington varnished wood model and the Teddington wax model. The wax model was afterwards coated with varnish and with red lead paint, but these coatings did not have any appreciable effect on the result.

One other point touched on by Mr. Robertson in connection with skin friction is the effect of fullness of form on skin friction. In the latest edition of my book a method

is suggested for making calculations to take account of this effect, but the C constants given in recent papers of the tank have been corrected without any such modification. Its effect is not so large as might appear at first sight since it is to be made both on model and ship friction.

In our reports of forms we work it out in some cases, and the total effect for large merchant ships is usually of the order of 4 to 6 per cent, and this percentage includes the effect of the adoption of the Teddington version of existing skin-friction data. One point of difference in Mr. Robertson's correction and that adopted at Teddington is the standard temperature that is taken at 70° at Washington and is 55° at Teddington, which would have the effect of making our results appear a little worse than those of Washington.

Thirdly, with regard to the "humps" which so often show on the C curves of ships. It is quite true that in some cases the humps due to the P values only are masked by other important developments, as suggested by Mr. Robertson. One of these important developments I have given in some detail in my book, namely, what is called an entrance wave. But when a hump occurs, it is very often quite easy to detect what it is due to, after the P values for the boat are worked out as suggested by Mr. Kent and myself. For example, in the figures published by Mr. Robertson there are no less than thirty humps, the majority of which can be explained by due and proper consideration of the P values at which they occur and the point at which the entrance or run will begin serious wave making in their own length. There are two or three cases in which we are not able to identify these humps, but that is all. I may illustrate the preceding by pointing out that the bad result obtained by model CX in Fig. 17, CW in Fig. 16, model CW in Fig. 11, model DV in Fig. 10 (Plate 23), and model BX in Fig. 5 (Plate 19), are all due to bad combinations of their P_3 or P_4 hump with the entrance hump. Although plotting to a P base does not always show humps, yet when they occur we have found at Teddington that such a plotting enables one to detect the cause of the hump with comparative ease.

Mr. Robertson (Communicated):—In reply to Mr. G. S. Baker, I am informed that shortly after the establishment of the Ann Arbor tank Dr. Sadler towed plane surfaces therein, the results of which when plotted differed so little from the resistances calculated by the Tideman coefficients that those coefficients were adopted as standard at Ann Arbor.

At Washington, in 1901, tests were made with a 20-foot plane of 64.1 square feet area, and the results were tabulated and are constantly in use in the estimation of wetted surface friction. The formulae in use at these two tanks are shown on Fig. 4, Plate 18. It will be noticed that the frictional resistance for the ship is based on the same formula for both tanks, the C difference being due wholly to the estimation of the frictional resistance of the 10-foot or 20-foot model. The introduction of a temperature correction to the Ann Arbor models is an innovation in the results herewith, the correction being similar to that used at the Washington tank.

I can see many reasons why a standard formula would be very desirable for all the tanks, both for temperature correction and for frictional resistance, and the first step towards this is to let some light into what the tanks are presently using. There is already a large body of information at each tank calculated by individual formulae, and conver-

sion tables would be necessary if a change is made, but this only makes it the more necessary to act at once.

I am glad to notice that Mr. Baker has developed further the effect of fullness of form on skin friction, and believe some such correction as he suggests is necessary. It must not, however, take the nature of a guess, or it would thereby largely vitiate the value of the tank tests to which this correction is applied.

Regarding the use of the P value, I would be very glad to have further information. Where the prismatic coefficient is the same, the P value, of course, is the same, and therefore for each of Figs. 15 to 20, Plate 23, the P would be identical for each curve. In Fig. 18, for example, P_4 is at about $57 \frac{V}{\sqrt{L}}$, and that gives us a hump on DX and a hollow on FV. It appears to me that the theory regarding the position of these humps is good, but is still incomplete.

The Chairman:—I am sure you will all desire me to express the thanks of the Society to Mr. Robertson for his valuable paper and the discussion which has resulted.

We will now pass to the next paper, the last one for this afternoon, No. 5, entitled "Notes on Rivets and Spacing of Rivets for Oil-Tight Work," by Mr. Hugo P. Frear, Member.

Mr. Frear presented the paper.

NOTES ON RIVETS AND SPACING OF RIVETS FOR OIL-TIGHT WORK.

By Hugo P. Frear, Esq., Member.

[Read at the twenty-eighth general meeting of the Society of Naval Architects and Marine Engineers, held in New York, November 11 and 12, 1920.]

It is improbable that the writer would have selected the above subject, which was assigned to him, had he volunteered to submit a paper at this session. The secretary usually sends out notices at an early date inviting members to recommend subjects and authors, and "Rivets and Spacing of Rivets for Oil-Tight Work" appears to be one of the topics which attracted the attention of the Council. An exhaustive treatment would be largely a matter of compilation and beyond the scope of this paper and will not be attempted, it being the desire rather to invite discussion if there is a difference of opinion regarding general practice or on the few points raised, without which the paper will be of comparatively little value.

The present high state of efficiency of the modern tanker leaves little to be said on rivets and rivet spacing for oil-tight work that would be both new and conservative, and it may be safely said that radical departures from the practice arrived at as the result of years of experience would invite trouble.

Riveting is unquestionably one of the greatest, if not the greatest, item of expense entering into the cost of construction, and the unprecedented demand for tankers and oil-burning vessels at this time magnifies its importance.

Experience has become much more general in both the building and operating of tankers, and perhaps this was thought to be an opportune time to provoke discussion along other than traditional lines.

Rivets and spacing of rivets for oil-tight work, as covered by the Classification Rules, Navy Department, British Admiralty, etc., do not vary radically although, on account of higher stresses used, a greater spacing is adopted in some parts for naval vessels. It must be borne in mind, however, that naval vessels are not subject to the great variations in load encountered in the merchant marine, and therefore the stresses can be anticipated with greater accuracy.

Rules for rivets and rivet spacing are the result of evolution arrived at by rule of thumb, experience, size of rivet found practicable to drive, spacing suitable for caulking, experiments, calculations, survey reports on damage cases, etc., and are in the opinion of the writer generally satisfactory with very few exceptions, which will be referred to later.

Oil tightness depends, perhaps, more on good workmanship than any other one thing, especially so far as riveting is concerned. If every rivet could be guaranteed 100 per cent perfect, slightly greater spacing would be practicable and might be allowed by classification societies except where a given efficiency of joint

is essential. The same standard of workmanship does not exist throughout the yards of the world or in some cases even throughout the same vessel. It is obvious that there must be a common rule for all and, therefore, the required rivet spacing is designed to guarantee good results for average standards of workmanship.

It was stated above that the rules for rivets and spacing of rivets for oil-tight work had been arrived at by a process of evolution, and it may not be out of place to refer briefly and in a very general way to some of the early bulk oil carriers and to some precedents in connection with conversion jobs on account of the large number of vessels which have been and will be converted to carry fuel oil.

While there were probably some barges in existence prior to 1872, it was in that year that the first bulk oil-carrying steamer was built at Palmer's, Newcastle on Tyne, England. Although this vessel was built especially for carrying oil in bulk, she had a double bottom, and the indications are that the rivet spacing was the same as in vogue at that time for water-tight work. Following this date, new bulk oil carriers were built from time to time, but perhaps for about three decades converted vessels contributed more or less to increasing the fleet.

During this period the spacing of rivets, even for water-tight work, was in some respects greater than at present and vessels frequently leaked a little when new, but they were built of iron and the joints soon rusted up tight, thereby increasing the friction of the riveted joints and resulting in stronger and more durable vessels under existing conditions. This rusting-up process, however, would not take place in the case of new vessels built to carry crude or lubricating oils in bulk, and on that account some of the converted vessels were tighter than new tankers. It was during this period that the importance of the friction of the joint referred to above became more generally appreciated.

Oxidation, however, takes place in tanks carrying gasoline, and this rusting-up process, undoubtedly, penetrates into the seams. On this account it might be beneficial to carry a cargo of this character on the first voyage or two as is sometimes done. Tests have demonstrated that, while the friction of the joint has no effect on the ultimate strength, it has a safe factor above working stresses where workmanship is good. If there is the slightest slip, defective rivets will come to light; therefore, in oil-tight work intimate metal-to-metal contact is of the greatest importance for permanent oil tightness, especially where subject to alternating stresses set up by the working of the ship.

Serious recurring damage, constant necessary repairs and resulting cost to underwriters during much of this part of the history of oil carrying in bulk were attributed to inadequate scantlings, improper distribution of material, insufficient riveting, poor workmanship, long tanks, overloading and limited knowledge of manipulating large quantities of water ballast, especially in the case of exceptionally long tanks. Investigation by Dr. James Montgomerie over a period of thirteen years prior to 1909 conclusively showed that recurring damage was more frequent where the tanks were over 30 feet long.

The early tankers, both new and converted, had relatively less beam and were consequently easier in a seaway, yet the penetrating character of the oil

sought out weak spots and loosened rivets developed under new stresses created by carrying oil in bulk.

The first leaks frequently occurred at the ends of stiffeners and in rivets connecting frames to shell which were subject to stresses not common in ordinary vessels. After these started the vessel began to work more, and progressive damage followed, frequently resulting in almost entirely reriveting the vessel.

There was a fad about this time for plug-headed rivets, but these were abandoned because they would not draw the work up sufficiently close and created greater difficulty in securing high quality of workmanship. The old reliable pan head was found to produce the most intimate contact and to be more permanently oil-tight and was recommended except where some other types were absolutely necessary.

Up to 1894 there were built or building to Lloyd's Class sixty-four tankers, yet the Society did not consider that it had sufficient experience to lay down definite rules for their construction. In this year, however, Mr. Benjamin Martell, chief surveyor to Lloyd's Register, read a paper before the Institution of Naval Architects wherein he outlined his views on the details of construction and of riveting for bulk oil carriers, recommending three diameters for oil-tight work. This paper in pamphlet form was distributed and for a number of years served as a guide to shipbuilders in the design and construction of tankers intended to be classed at Lloyd's.

Within a comparatively short time after this, Bureau Veritas, apparently profiting by this paper, published rules for tankers, and the American Bureau of Shipping published their first rules in 1902, but the first rules of Lloyd's did not appear for general use until 1909.

Both Bureau Veritas and the American Bureau recommended three and one-half diameters at the start, while Lloyd's specified one more rivet in a frame space than required for water-tight spacing resulting approximately in three and one-half diameters.

According to the recollection of the writer there was a period when Lloyd's practically decided not to class any more converted vessels, but both workmanship and details of construction must have improved, as many converted vessels have since been classed and have given good service.

Evolution was a slow process, and no doubt in numerous cases the true cause for failure was not recognized. Mr. Martell realized this when he wrote his paper and provided for tanks not over 24 feet in length, increased stiffeners and brackets, as well as closer spacing of rivets. Had the earlier tankers been better designed as regards tanks, stiffeners, brackets and compensation, where needed, the question of rivets and spacing of rivets for oil-tight work might possibly have received somewhat less attention during those troublous times. This appears to be a reasonable inference from the fact that so many converted tankers have since given long and satisfactory service.

Rivets in the shells of the early tankers were continually getting loose and leaky, while it is very rare that the shell rivets spaced for water-tight work of later

converted vessels, where the interior details of construction were in accord with Mr. Martell's recommendations or classification rules, showed leaks. The comparatively slight variation in the spacing of shell rivets would not account for this great difference and must be attributed somewhat to improved details of construction and, in a measure, superior workmanship.

That oil-tight work should be double riveted was an accepted rule. This is still the general requirement, but single riveting is now permitted in the seams of double bottoms intended for carrying fuel oil. This concession was not made until after many vessels in commission had been converted to carry fuel oil in their double bottoms with single-riveted seams and their class continued. The writer recalls a large vessel on the stocks with tank top required to be double riveted to carry fuel oil, while at the same time an old vessel lying in a wet slip alongside was being fitted to carry fuel oil in the double bottom where the tank top was single riveted.

Both new and old single-riveted tank tops, where fuel oil is carried in double bottoms, have proved a success. Why should there be any greater risk in single-riveting the seams for decks and trunks of tankers, which are subject to less test head and deflection, especially where the spacing is made closer for oil-tight work? This is a parallel case to double bottoms above noted, for both Lloyd's and the American Bureau have vessels on their books which have been fitted with deep tanks for fuel oil, retaining the original single-riveted decking. This point has been discussed with representatives of both the above societies and one of the largest shipowning oil companies who agreed that there would be no real objection with crude oil.

Just how far this argument could be carried for tankers intended or liable to carry bulk gasoline or other light distillates is an open question. If single-riveted seams are tight for crude oil, would they be as tight or rust up tighter with gasoline? Lloyd's do not permit gasoline to be carried in single-riveted double bottoms, but it is not known whether or not it would be permitted even with double-riveted seams with dry cargo above.

The writer was closely associated with the conversion of three passenger steamers to carry about 18,000 barrels of fuel oil in deep tanks forward of the fire room. The vessels had double bottoms which were opened so as to form part of the deep tanks, but the water-tight divisions in the double bottom did not coincide with the tank bulkheads and a portion of the double bottom under each tank formed a part of an adjoining deep tank. In other words, the transverse bulkheads extended, so to speak, under the next tank to the first double-bottom division and then down to the keel, the new part above double bottom being double riveted and the horizontal part, or what was the original double bottom, being single riveted. The after tank extended in this way under the forward fire room. These vessels have given excellent service, notwithstanding they were built during a strike where the best work could not be expected.

It is not concluded from this that it would be advisable to single-rivet transverse bulkheads in tankers, but, with a view to eliminating possible unnecessary

expense, it might be well worth considering whether or not there would be serious objection to single-riveting the longitudinal seams of center-line bulkheads with the exception of the lower seam. A cargo is sometimes made up of more than one kind of oil, and it is therefore important that the transverse bulkheads be absolutely tight, but different kinds of oil have seldom been carried on opposite sides of the center line, and it would appear that a single-riveted longitudinal bulkhead would fulfil every function the trade demands.

As noted above, Lloyd's in their first announcement, more or less official, recommended three diameters for all oil-tight spacing. This was, in some cases, applied literally. Two vessels were converted in the vicinity of New York to carry oil in bulk, on which a spacing of three diameters was required in both rows of rivets in the flanges of single 5 inches by 5 inches bulkhead bounding bars, which both crowded the rivets and unnecessarily impaired the strength of the shell plating. Classification societies now allow a spacing of five diameters connecting bulkhead bounding bars to shell.

Lloyd's formerly recommended single double-riveted bulkhead bounding bars but later gave the builder the option of using double single-riveted bars. The general practice now is to use double single-riveted bars with countersunk heads and points and bars caulked on both flanges on each side. The writer is inclined to favor single double-riveted bounding bars and pan-head rivets in standing flange with caulking on one side because it is believed this lends itself to cheaper and better workmanship and would insure the parts being drawn up more closely. Countersunk heads are, of course, unavoidable in the shell flange.

On the Clyde, bulkhead bounding bars are frequently hydraulically riveted on the ground, using pan-head rivets and finishing with snap heads and points. Similar practice is universally followed in this country in the case of locomotive oil tenders and tank cars. Snap rivets driven by hand are also satisfactorily used in the construction of oil storage tanks.

There are two vessels now building for the same owner in different yards with double single-riveted bulkhead bounding bars with countersunk head rivets where in one case every rivet is required to be caulked head and point before testing, while in the other case none of the rivets are being caulked before testing. A good pan-head rivet should never require caulking before testing. Pan-head rivets are used in the bulkhead stiffeners and are not required to be caulked before testing, which seems to indicate that it is considered the countersunk head rivet requires more doctoring.

In the case of another vessel now building the inspectors are instructed that rivets are to be caulked before testing, in bulkhead bounding bars, heads and points, in all four flanges, also rivet points through bulkheads in clips connecting longitudinal brackets to bulkheads, while in the same instructions it is recommended that as few rivets as possible be caulked before the water test, as it is desired that the rivets be driven in such a manner as to be tight without caulking.

Where double bounding bars are fitted, careful workmanship is necessary to insure the two shell flanges lining up perfectly for intimate contact. To overcome

this difficulty and with a view to better and more economical work, T-bars have in some cases been used, although the present available 6½ inch by 6½-inch T-section does not provide sufficient width of shell flange on that side on which the bulkhead plating connects to the standing flange for best spacing of the rivets. A more general use of this type of bounding bars has been recommended by some owners, and with this in view two of the large shipbuilding concerns have without success urged the steel companies to roll a larger section. Conditions, however, would be substantially the same as with single bounding angles. Pan-head rivets could be used except in one flange, three-ply work would be eliminated, and only two edges would require caulking. A single angle as recommended above would have the same number of rivets, the same amount of caulking, and be more easily set.

In gunwale bars, Lloyd's require four and one-half diameters for water-tight work and five diameters for oil-tight work, which does not seem consistent. In a recent case, 1-inch rivets were spaced five diameters in a 5-inch by 5-inch gunwale bar for oil-tight work and appeared to be too crowded for the width of flange and size of rivets. Probably a larger bar or six diameters would have presented a better appearance.

The table on the following page gives, for a number of selected parts, the rivet spacing recommended by Mr. Martell in 1894 for oil-tight work and present requirements for both oil and water-tight work of Lloyd's, American Bureau, United States Navy, and British Admiralty.

Conditions briefly referred to in the table, no doubt, justified Mr. Martell in recommending a spacing of three diameters for satisfactory oil-tight work, but better design, construction, and workmanship have demonstrated that strong, staunch ships can be built with wider spacing and be suitable for carrying oil in bulk. Scantlings and riveting must be sufficient to resist the stresses brought upon them, and there must be no slipping of the joints. When this condition is fulfilled the work should be oil-tight whether the spacing is according to present oil or water-tight practice. It will be noted in connection with merchant vessels converted to carry cargo or fuel oil that the spacing is generally four and one-half diameters in the shell.

If oil-tight work depended less on the strength of the riveted joints and of the structure as a whole, and to a greater extent on spacing that would permit caulking without the edge raising, wider spacing could be used. Work that might be tight under static head would not necessarily be tight on parts subject to the working stresses of a vessel in a seaway, but might on certain parts where these stresses were in a measure absent. Any design, spacing, or workmanship which will not permanently prevent the joints from slipping will not remain oil-tight.

The extensive use of oil for fuel in naval vessels during the last few years has brought up for consideration and settlement many problems in design which did not exist in coal-burning vessels. In vessels of great length such as scout cruisers and battle cruisers, longitudinal bulkheads are used as strength members. The riveting connections in these bulkheads must be designed to obtain maximum

	Martell, 1894	Lloyd's 1919-20 rules		American Bureau 1919-20 rules		U. S. Navy		British Admiralty	
	O. T.	O. T.	W. T.	O. T.	W. T.	O. T.	W. T.	O. T.	W. T.
Shell plating seams	3	3½	4	3¾	4½	3½	3½–5	4	4–4½
Deck plating seams	3	3½	4½	3¾	4½	3½	3½–5	4	4–4½
Bulkhead plating seams	3	3½	4½	3¾	4½	3½	3½–5	4	4–4½
D. B. plating seams	3	3½	4	3¾	4½	3½	3½–5	4	4–4½
Tunnel plating	3	3½	4½	3¾	4½	..	..	4	4–4½
Frames to shell	6	6	7	6	*	8	..	4	7–7½
Deck bars	..	5	4½	..	..	..	4½	4	4½
Bulkhead bounding bars	3	5	5	5	5	5	5½	4	4½
Bulkhead stiffeners	6	6	7	†	†	6	6	6	7
Tunnel stiffeners	6	6	7	7	7	..	..	6	8
Shell butts	3	3½	3½	3¾	4½	3½	4½	4½	4½–5
Deck butts	3	3½	4	3¾	4½	3½	4½	4½	4½–5
Bulkhead butts	3	3½	4½	3¾	4½	3½	4½	4	4½–5
D. B. butts	3	3½	4	3¾	4½	3½	4½	4	4½

* 7 up to 27-inch frame spacing.
6 above 27-inch frame spacing.
5½ inch deep tanks.

† 7 generally.
5½ deep tanks.
4 for 15 per cent length each end.

strength, and therefore the riveting should not be so closely spaced that more material is punched out of the plating than allowed in the calculations, without compensation. If, however, these bulkheads form boundaries of oil tanks, it becomes a matter of importance, when transverse bulkheads join the longitudinal strength bulkheads, to avoid too close spacing in the bounding bars. If the strength bulkhead is designed for a spacing of six diameters for rivets in stiffeners and oil tightness demands five diameters at a transverse bulkhead, it becomes necessary to introduce compensating liners which add to the weight and, on account of the extra thickness of plating, increase the difficulty of making the structure tight.

The Hull Division of the Philadelphia Navy Yard recently made a test on an experimental box to determine if a six-diameter spacing of rivets would be satisfactory under the conditions as stated above. It may naturally be expected that a wider spacing for oil-tight work can be used where the bulkheads are close together, where the stresses due to swash are not severe, and where alternating stresses are of secondary consideration. Six diameters might be quite suitable on a deck near the neutral axis but unsuitable for parts subject to stresses resulting from the vessel working in a seaway or from shocks of gun-fire or explosion. In this test it was found that the caulking of bounding bars to heavy bulkheads was satisfactory under a 60-foot head with a six-diameter spacing in the connection of

transverse to longitudinal bulkhead and five-diameter spacing in bar connecting to the shell. No gunning was used and no special effort made to have the box absolutely tight on the first test.

A similar tank was also recently tested at the New York Navy Yard to demonstrate the efficiency of a six-diameter spacing for T-bar stiffeners and for bounding bars to bulkheads. The tank was first subjected to a test head of 65 feet of water and found to be tight; then to a twenty-four-hour endurance test with oil at the same pressure. Then the pressure head was increased to 93 feet, which pressure was maintained for four hours without leaks.

The results of the tests appeared to indicate that in fairly heavy plating, where the members are not under the influence of other stresses than those due to head, six diameters are suitable. The thickness of angle bars, it was suggested, should preferably be not less than three-eighths of an inch.

It is understood that further tests were directed to be made on experimental tanks at navy yards, as the Navy contemplated revising its rules for oil-tight riveting due to the very considerable increase in amount of oil-tight work involved by new construction. A description of these tests in greater detail would be interesting, but, unfortunately, full particulars are not at hand.

It has been the practice in naval work to use a spacing of three and one-half diameters for oil-tight work in seams and straps, four diameters in angles to plates including staples, and four and one-half diameters on bulkhead stiffeners if caulked oil-tight. This is a closer spacing than is required for water-tight work. Experience in the construction of oil tanks in double bottoms seems to indicate that, with ordinary good workmanship, water-tight spacing will be satisfactory for oil. While the naval authorities are not prepared to make the spacings the same, a compromise has been made allowing a somewhat greater spacing than called for by the navy rules for oil-tight work. For single riveting three and one-half diameters are retained, but four and four and one-half diameters, depending on the thickness of the plates, are now accepted; five diameters are used in bounding bars of oil-tight floors.

One of the larger builders, as a result of mathematical analysis, has suggested the following spacing as permissible, provided that considerations other than oil-tightness did not determine the spacing:—

Diameter of rivet	*Thickness of plating*	*Spacing in diameters*
3/4″	5/16″	3.6
3/4″	3/8″	4.2
7/8″	1/2″	4.9
1″	3/4″	5.6
1 1/8″	1″	6.3
1 1/4″	1 1/4″	7.0

The above spacings were based on the supposition that three and one-half inch diameters were satisfactory for a 10-pound plate using five-eighths-inch rivets, and

assuming that the deflection of the plate edge caused by caulking probably varies as the ratio of the fourth power of the distance between rivets to the fourth power of the thickness of the plating, and that the extension of the rivet varies as the ratio of the distance between rivets to the area of the rivet. In order to make practical use of the above it was suggested that water-tight spacing be used for rivets in oil-tight work on naval vessels.

A naval constructor stationed at the same yard carried the investigation further by considering the deflection that is caused by caulking and by the amount of general deflection of plating due to water pressure, and suggested that based on the assumptions made the following spacing would be satisfactory:—15-pound plate, 4 diameters; 20-pound plate, 4.5 diameters; 25-pound plate, 5 diameters; 30-pound plate, 5.3 diameters; 35-pound plate, 5.6 diameters.

These conclusions were deduced without reference to strength of joint, and it was pointed out that this consideration might be more important than questions of tightness. For five-eighths-inch rivets in 10-pound plating it was shown that approximately .006 inch is the deflection or amount of seam that can be caulked off without exceeding the elastic limit of the material. The stress on the rivet for such caulking was found to be very moderate.

The analysis used in deriving the above tables of rivet spacing for oil-tight work was made to confirm a theory that every unnecessary rivet is a possible source of leakage, and that for heavy plating a wider spacing would be permissible but considered only the stresses on the plating and rivets due to caulking to the exclusion of the more complex stresses experienced under service conditions.

For thin plating, where large rivets in relation to thickness are used, the diameter is based more nearly upon theoretical considerations, but if the same considerations were followed in determining the diameters for thicker plates, the rivets would exceed the limit practicable to drive, and spacing based even on using the same number of diameters would at some point become greater than indicated in the tables. Thick plates, however, are used where greater strength is required, and as the rivets used are much smaller in relation to thickness a greater number is necessary to develop the strength of the plating. The uniform spacing in diameters adopted in practice, substantially, for all thicknesses of plating is a great convenience, and it is only necessary to increase the number of rows of rivets where greater efficiency of joint is desired. If the spacing suggested in the tables given is suitable for oil-tight work, it might be necessary to increase the number of rows to insure proper strength and friction of the joint, in which event there would be an increase in weight and the attempt to reduce the number of rivets defeated.

DISCUSSION.

THE CHAIRMAN:—Gentlemen, this paper, No. 5, entitled "Notes on Rivets and Spacing of Rivets for Oil-Tight Work," is now open for discussion.

MR. E. H. RIGG, *Member:*—This subject is a live one at the present time. Oil tankers and oil-bunkered ships are in demand, and their economical construction is one of the problems exercising the staffs of all our large yards and of some yards not so large.

The troubles in the coal mines all over the world, the ease of handling oil at the ports and in bunkering ships and the greater thermal value of oil all tend vigorously to increase its use—a point which was brought out more effectively by Mr. Ferguson this morning than I can hope to bring it out now.

Our modern merchant marine is overwhelmingly oil-burning—our modern navy entirely so.

The problems connected with oil-tight riveting first comes to a head at the ceremony of testing tanks; it will be more or less of a ceremony depending on the excellence or otherwise of a series of operations starting before template making and ending with riveting and caulking. Service conditions have their say later.

Mr. Frear's paper bears evidence of considerable work. I feel somewhat diffident in discussing it, having been on the office end of our profession so long that yard problems are not as fresh in mind as I would like them to be. In talking it over with some of our (New York Shipbuilding Corporation) outside people the following points were suggested, mostly by them, as being worthy of consideration:

1. Referring to double versus single bounding bars; in transversely framed ships the extra rigidity given by the double bar was worth the extra fitting; in longitudinally framed ships, however, the rigidity imparted by the longitudinal frame brackets makes the single bar the preferable job, the two-ply riveting advantage being thus obtained.

2. Another suggestion was made to the effect that for oil work a pan-headed rivet with a slight collar formed on the outside at the base of the head would be a good thing, so as to enable caulking to be done if found necessary on testing. There is enough demand for these rivets to justify special dies for the head.

3. In many vessels, merchant and naval, we have bunker tanks that are quite deep and which consequently demand the best workmanship to ensure tightness. Our naval constructors are entitled to credit for their efforts to successfully reconcile the spacing for strength with that for tightness. These efforts result generally in wider spacing, giving greater strength, less riveting and at the same time satisfactory conditions as to tightness. British practice seems to bear this out.

4. Single-riveted seams for center-line bulkheads referred to on page 65 of the paper are something that will be admitted only after considerable discussion. There will be times in port when a considerable head will be on the bulkhead with the vessel loading and unloading; the upper and lower strakes at least are integral members of the longitudinal girder requiring rigid connection to the adjacent plates. Also during testing of tanks, center-line bulkheads will get heavy unilateral loading.

5. All efforts to economize and keep down production costs are to the point in these days, and it would be interesting to hear from the American Bureau as to their experience with single-riveted seams in the upper parts of oil-tight bulkheads as these are apparently permitted under their rules (Section 30, sub-section 20, page 88).

6. The satisfactoriness, or otherwise, of single-riveted seams in double-bottom tank tops is largely a matter of height of test head. The tank top is naturally a stiffer structure than decks or bulkheads, but when you get a 40 or 45-foot test head it is well to pause before adopting single-riveted seams.

COMMANDER A. H. VAN KEUREN, C. C., U. S. N., *Member:*—When the advance copy of this paper was received at the navy yard, New York, it was noted that mention is made of tests conducted at the yard, as well as of navy practice in general. It is thought that a brief statement as to these tests and as to present oil-tight riveting practice on naval vessels may be of general interest.

The first test, as mentioned by Mr. Frear, was made on a tank representing a section of adjacent torpedo bulkheads stiffened by T bars and separated by transverse oil-tight bulkheads. This test fully demonstrated the efficiency of six-diameter spacing for T-bar stiffeners and for bounding bars to the torpedo bulkheads, the pressure head running up to 93 feet. Naturally the five-diameter spacing in bounding bars to the transverse oil-tight bulkheads stood up under the same pressure head.

The further test alluded to by Mr. Frear was made on a tank built up of four thicknesses of plating, namely, 12, 15, 20, and 25 pounds, respectively, on the four sides, and with 15-pound and 25-pound ends. A 20-pound bulkhead divided the tank into two sections, A and B. Each side had two seams, one lapped, the other strapped. The 12-pound plating had a single seam strap double riveted in both A and B sections; the 15-pound plating had a single strap double riveted in the B section and a double strap in the A section; the 20-pound and 25-pound plating had both single and double straps similar to the 15-pound plating. The bounding bars were mitred in two corners of each end of the divisional bulkhead and electric welded in place. Double bounding bars were used but stapled on one side only. The rivets were only caulked as necessary to take up leaks on the first test. The rivet spacing was as follows:

CONNECTIONS.	RIVETING.	DIAS.
Plating: Straps and butt laps	Under 14 pounds	3½
	Over 14 pounds to 20 pounds	4
	20 pounds and over	4½
Double butt straps	Under 14 pounds. (Straps 9 pounds)	3½
	Over 14 pounds to 20 pounds. (Ck. strap 15 pounds; inside strap 10 pounds)	4
	Or over, 20 pounds or over. (Ck. strap 20 pounds; inside strap 15 pounds)	4½
Seam laps and seam straps	Under 14 pounds	3½
	Or over, 14 pounds to 20 pounds	4
	Or over, 20 pounds or over	4½
Angles to plates, including staples	Under 15 pounds	4
	15 pounds or over	5
	15 pounds or over	5½

Rivet holes were punched small and reamed. All rivets were pan head and countersunk point except in bounding bars of ends and division plate where they were countersunk head and point.

In testing this tank it was desired to obtain a comparison between the tightness when under pressure with water, air, fuel oil and gasoline, respectively, no work

being done on the tank between the various tests. Preliminary tests were carried out in the order named, with pressures up to 30 pounds, and the leakages and drops in air pressure were measured as accurately as possible under various pressures for equal lengths of time. The tank as a whole showed up well under the various tests with the exception of the single-strapped seam in the 12-pound plating which began to leak at 10 pounds pressure through the butt caulking of the seam. The single seam in the 15-pound plating also showed leakage at a slightly higher pressure. This made it difficult to compare the leakages under the various mediums, but, generally speaking, the leakage in section A, which was slightly less than in section B, because of the double seam strap in the 15-pound plating, showed up greater for oil and gasoline than for water, the gasoline being the greatest. The leakage of air was easily detected by using soapy water, and as a medium for testing tightness it is considered that air, as has been frequently shown in compartment tests, is fully as searching as water, oil or gasoline. The greater leakage of gasoline as compared with oil may be partly attributable to the gasoline test having been conducted after the oil test. One result was particularly noted; namely, that in the oil test under a 65-foot head the leakage was half that for oil under a 45-foot head, being 7 pints as against 14 pints. This is probably due to the greater deflection closing up the seam. The electric welding showed no leaks in any of the tests, nor did the riveting as a whole, only a few weepy rivets being noted. There was some slight leakage through caulking of bounding bars.

After the preliminary tests it was decided to renew the 12-pound plate, using a double-riveted lap instead of a single strap, double riveted, the rivet spacing remaining the same. The tank was then again tested with water, air, oil and gasoline, using 30 pounds of pressure as the maximum. Both compartments were perfectly tight with water, air and oil. When gasoline was applied to section A, after 30 pounds of pressure had been on for an hour, a slight leak developed in the bounding bar to 15-pound plating. After 30 pounds of pressure of gasoline had been on section B for three hours a bad leak developed in the seam strap of the 15-pound plating, which showed up in the inner strap of section A; also at the corners of the bounding bar, the total amounting to 3½ gallons in six hours.

The marked improvement in the behavior of the lap over that of the single-riveted strap is worthy of note. It is regretted that a single-riveted lap was not subjected to test also.

Commenting on that portion of Mr. Frear's paper dealing with naval practice and the present tendency to allow a somewhat greater spacing than called for by the navy rules for oil-tight work, it may be stated that for some at least of the naval vessels building, the following oil-tight practice is observed generally (see table, page 73):

This practice is used as a general guide, border-line cases being settled on their merits or on special considerations of strength.

On battleship work it is customary to fit double bounding bars on oil-tights between torpedo bulkheads, double-riveted 5 diameters through the transverse, and single riveted 6 diameters through the torpedo bulkheads, both bars being stapled. In service oil tanks in double bottoms, double bars, single-riveted 5 diameters are used, both stapled. In emergency oil tanks in bottoms (reserve feed and void spaces), double bars, single-riveted, 5 diameters are used, one bar only being stapled.

As regards caulking of oil-tight rivets before testing, the yard endeavors to restrict this to a minimum.

CONNECTIONS.	READING.	DIAS.
Angles to plates, including staples.	Under 15 pounds	4
	15 pounds and over	5
	Under 15 pounds	5
	15 pounds and over	5½
Seam laps and seam straps		3½
	Under 14 pounds	3½
	Or over, 14 pounds to 20 pounds excl	4
	Or over, 20 pounds and over	4½
Single butt straps and butt laps		3½
	Under	3½
	Or over, 14 pounds to 20 pounds excl	4
	Or over, 20 pounds and over	4½
Double butt straps		3½
	Under 14 pounds	3½
	Or over, 14 pounds to 20 pounds excl	4
	Or over, 20 pounds and over	4½

With reference to the use of T bars no special difficulties have been encountered as yet, although the design does not call for their use to replace double bounding bars. The only riveting troubles have been due to bevels in the T rather than to spacing of rivets.

Mr. Harold P. Norton, *Member:*—Mr. Frear's paper is a very excellent one, because it points out most of the matters under discussion and brings out all of the arguments pro and con, but makes very few attempts at definite recommendations.

It has been our experience at Newport News, in connection with oil-tight work, that perhaps as much depends upon the riveting and caulking conditions as upon theoretical considerations. For that reason we are a little inclined to favor the double bounding bars for bulkheads, but we do believe that it is only necessary to have the rivets countersunk, head and point, in one of these bars. We believe that the panhead rivet on the rough side of the bulkhead makes the best job, because the panhead rivet draws up the work better and gives a tighter joint. The reason we favor the double bar is because the bar on the other side backs up the caulking of the bar on the smooth side, and it seems to us that therein lies very largely the secret of success of oil-tight work—to see that all of the caulking is properly backed up. That is largely the reason why in naval work six-diameter spacing is considered sufficient in a bar at least a half-inch thick, against a good, thick bulkhead, such as a longitudinal torpedo protection bulkhead, and why a five-diameter spacing is used for the lighter plating of the adjacent transverse bulkheads.

It seems to us that it would be to the advantage of the classification societies to examine this class of work of the Navy Department, in connection with which the investigations mentioned in the paper were made, with a view of perhaps permitting some slightly wider spacing in connection with certain merchant work.

The matter of the double-riveted seam is one in which there seems to be considerable difference of opinion, and we suggest that the principal advantage of the double riveting is in the bolting up of the work. The work can be bolted up better with the double-riveted lap than with the single-riveted lap, and perhaps that is the secret of the advantage of the latter. But right in this connection there is another possibility, and that is

that perhaps the real way to make double-riveted laps is to close space the rivets on the caulking edge and wide space those on the rough edge. That would give the advantage of a double-riveted lap in bolting up and at the same time save a large number of rivets.

Mr. Robert W. Morrell, *Member:*—I have not had a chance to study Mr. Frear's paper, so I cannot discuss it fully, but two or three things occurred to me as I read it over.

With respect to the double bars against the T bars, we are inclined to favor the double angles, for the reason that Mr. Norton mentioned, but we do think there is some advantage in the T bar, due to the fact that there is somewhat of a vacant space between the heels of the double bars in which the oil or water or rust will collect, and we have known cases of contamination of cargo due to dirty oil which lodged in those spaces, entering the tanks after they were loaded. It is true that there is a certain amount of oil which collects in those spaces. That is one reason why we favor the use of countersunk heads in the bounding bars on the rough side of the bulkhead.

With reference to the caulking of the rivets, we are strongly in favor of driving rivets that do not need to be caulked, and we believe that if it is known by the workmen that the rivets are going to be caulked, that they will not be so careful about them as they would if made to understand that the rivet is to be a good, solid job without caulking.

We favor, however, the caulking of the rivets in both flanges of the bounding angles and both sides of the bulkheads, because it is difficult to get at these places after the bulkheads are in the ship and the brackets are in place, and therefore it has seemed preferable to caulk them while we have an opportunity to get at them.

But in these matters it is very difficult to make any rules, because conditions vary with the different yards, with the class of workmen available therein, and we must be governed by the conditions which exist on the spot. In the same way it is rather difficult to make any rules applying to common, straight-away work, as opposed to places where it is difficult to get at them, such as corner staples, and a general rule which might apply to conditions in one place would not apply in another place.

In regard to the center-line bulkheads, it is true in a general way that we can carry the same cargo on both sides of the center line, so it is not usually necessary to have the absolute tightness there that we have in the transverse bulkheads. That is the general condition, but it frequently occurs that we carry fuel oil in half a tank on one side of the center line and cargo on the other side, in cases where we have to take a large quantity of fuel to provide for the return voyage. It has become necessary to do that in several cases, and under such circumstances it is essential that the center-line bulkhead be absolutely oil-tight. For that reason we would hesitate considerably in making a single-riveted seam in the center-line bulkhead.

I am inclined to believe that a revision of the rules might be towards a diminution of the number of rivets required, but that, again, is a matter which requires very careful thought, and we may not apply it to all situations occurring in the ship. It might apply to certain straight-away conditions, but might have to be modified to meet the particular conditions that would be met in the various members.

Mr. J. W. Stewart, *Member:*—I have read with interest the paper written by Mr. Hugo P. Frear, which opens up an important subject to shipbuilders today, and I compliment him on his methods of treating a peculiarly difficult subject.

I note he suggests that it might be beneficial to carry a cargo of gasoline on the first one or two voyages, so that what he terms the "rusting-up process" would take place and penetrate into the seams. In practice, I am afraid that this would be well-nigh impossible, particularly at the present time, when vessels are being used almost exclusively to carry crude oil.

In his paper he touches on excessive length of tanks, and he pointed out that information derived by Lloyd's Register of Shipping conclusively showed that recurring damage was more frequent when the tanks were over 30 feet long. The length of tanks, I consider, could go beyond that, provided precautions were taken in the upper portion of the tanks by extra stiffening on the bulkheads and the fitting of an intermediate swash plate. Extra precautions would also require to be taken in the riveting, or additional riveting might be required. Undoubtedly the reason why double riveting has been maintained by classification societies, particularly Lloyd's, is due to excessive swash in a seaway, and I think this is reasonable.

I do not think Lloyd's have refused conversions except in isolated cases, which was probably due to their desire to maintain a high standard at a time when little experience had been gained in the case of ships carrying petroleum in bulk. However, it was found that there was business for a large amount of tonnage in carrying heavy oil only, and they gave a class, "Carrying oil fuel in bulk, F. P. above 150° F." This class covered the conversion type and has shown that longer tanks and single-riveted center-line bulkheads, single-riveted decks and also single-riveted tank tops have proved reasonably efficient for a vessel of this class with oil which is generally crude and highly viscous. Also, such a proposition is already approved by Lloyd's in giving certain tankers the class "Carrying homogeneous cargo of petroleum in bulk," in which case the intermediate and center-line bulkheads are single riveted, the end and cofferdam bulkheads being double riveted.

The limitation to the length of tanks was more the result of Suez Canal regulations than anything else, on account of the danger of fire taking place during passage through the canal.

Mr. Frear mentions that double riveting in oil-tight work is now the accepted rule, and, as I have said before, I think wisely so, particularly as the tendency is to have longer tanks.

With regard to the rivet spacing, I agree with Mr. Frear that workmanship has a great deal to do with the present standards, and if the classification societies (now that air tools are in such universal use) would demand subpunching—that is, holes punched a size smaller, countersunk to suit the rivet required, and afterwards reamed out to the actual size, thus ensuring perfect holes, which incidentally would take away any burr on the faying surface of the plate—this, together with efficient bolting up at about every third hole, I think they could safely reconsider the question of spacing, particularly any portion where strength connections are not the feature. Single riveting, too, might even be agreed to in parts. I am sure the extra cost of subpunching would be more than offset in the labor saved in tank testing.

It may be, now that electric welding is becoming more reliable, and providing that one can ensure oil tightness, that single riveting might be more generally adopted with welding along one edge.

With regard to Mr. Frear's remark as to whether classification societies would permit dry cargo on top of double bottoms carrying gasoline, even if the seams were double

riveted, I think the risk would be too great, particularly in view of many disasters that have taken place, due to accumulation of gas with low-flash oil.

With reference to the single riveting of the seams of the center-line bulkheads, I might say that in vessels designed for carrying heavy oil the seams of the center-line bulkheads have in some special cases been allowed to be single riveted. If, as Mr. Frear seems to think, the same grade of oil is always carried on the opposite sides of the center-line bulkhead, and if the owner would agree to guarantee the same, I think the classification societies might be willing to consider single-riveted seams. In what was known as the "heavy oiler class," built during the war, and which has proved successful, quite a large amount of the work, such as the seams of the center-line bulkhead and some of the transverse bulkheads, the decks, and the tank top were all single riveted.

With regard to the caulking of rivets, I consider this should only be done where necessary, to save trouble in awkward corners. If care is taken by subpunching and close bolting, the caulking of rivets would be reduced to a minimum and need generally only be resorted to after testing.

I have observed men caulking rivets and undoubtedly making them worse, because where the point is unusually full and spreads over the surface, they very often caulk the plate away from the rivet, the result being a tendency to loosen the rivet.

I might say here that the practice which is very often adopted, of double-hammering the rivet, appears to me to be very good, and, if flat-headed hammers were always used, this would ensure the rivet filling the hole thoroughly, and caulking would be cut down to the minimum. Further, I agree with Mr. Frear that a good panheaded rivet should never require caulking before testing.

There is no doubt that the T bars are a boon to good oil-tight work, and it is a pity that there is not a slightly larger section available. It has occurred to me that a 7 inch by 7 inch angle, if rolled, would be very suitable and would take the place of the T bar, by fitting the plate in the bosom of the bar. I also agree with Mr. Frear that a 5 inch by 5 inch gunwale bar is unsatisfactory for oil-tight work and that a 6 inch by 6 inch bar is more suitable.

With reference to the spacing of rivets for oil-tight work, the unanimity of the requirements of the classification societies goes to show that, unless the work could be ensured by subpunching, etc., their tables appear to be reasonable. If the spacing should be modified in future, the thickness of the plates will be an important factor, as there is no doubt that in way of thicker plating (providing strength considerations allow for the change) wider spacing could be adopted, particularly starting above ½ inch thick.

Thought should be given to the riveting in the ends of stiffeners, and these should receive very special consideration, as experience shows that this is where trouble usually develops. This is featured in the Isherwood system by the close spacing at bulkheads and transverses, and Lloyd's rules require the riveting to be closed up to four diameters for 15 per cent at each end of unbracketed vertical stiffeners.

Rivet points should be maintained full and raised above the surface of the plating about ⅛ inch. On the other hand, there is a tendency in so doing to make the points unnecessarily full, which is a bad feature, as the rivet spreads over the plate and does not ensure good workmanship. Although it may look satisfactory, if the hole is not entirely filled, when any working takes place the rivet may become slack, where originally, to all appearances, the rivet looked good.

In conclusion, I might remark that undoubtedly these notes of Mr. Frear's will form a valuable addition to the Society's collection of data for the future.

Captain Robert Stocker, C. C., U. S. N., *Member* (Communicated):—The paper "Notes on Rivets and Spacing of Rivets for Oil-tight Work," by Mr. Hugo P. Frear, has been read with a great deal of interest, especially as the Bureau of Construction and Repair has given considerable thought and attention to this subject during the past four years; also as the types of naval construction for capital ships now make it particularly desirable to use wider spacing for oil-tight riveting. It is thought that some notes on the activities of the bureau in regard to oil-tight riveting may be of interest to the Society. The notes relate to oil-tight riveting, use of double versus single butt straps, painting of faying surfaces and oil stops.

The bureau's standard practice for spacing of rivets in oil-tight work, as contained in its "Specifications for Riveting," limits the maximum spacing of rivets in seam laps and seam straps to 3½ diameters, regardless of thickness of plating; also in butt connections 3½ diameters is the maximum spacing allowed, unless otherwise approved or directed. This specification limits the spacing in angles to 4 diameters, unless otherwise approved, and to 4½ diameters in bulkhead stiffeners, if caulked. The 4-diameter spacing in angles necessitates fitting compensating liners to strength members to which oil-tight structure is connected. Aside from the weight involved it is very desirable to eliminate compensating liners wherever practicable, because the fitting of the same requires an additional thickness of plating to be riveted. Each thickness means that much more difficulty in obtaining satisfactory work.

On the Colorado class of battleships and subsequent capital ships, the compartments formed by the torpedo bulkheads are fuel-oil tanks which requires this structure to be oil-tight. The spaces between the longitudinal torpedo bulkheads are divided by transverse bulkheads at intervals of about 24 feet into fore and aft tanks. The longitudinal bulkheads are stiffened at each frame on which a transverse bulkhead is not fitted by vertical web plates. Aside from the longitudinal strength that the torpedo bulkheads contribute to the ship, they must be capable of deflecting like a disc before rupture occurs, in order to accomplish the full purpose for which they are designed. It is therefore paramount that there be no marked line of weakness nor weak spot where a tear might readily start. As mentioned above, all work in connection with these bulkheads is required to be oil-tight and to compensate for the lines of weakness, caused by the close spacing of rivets connecting the vertical members to the torpedo bulkheads, compensating liners are required at each frame, each being about 30 feet in length amidships. It is obvious that, in a vessel as long as a modern battleship, this would require a large amount of bulkhead liners. It also means four-ply riveting in most cases. In laying out these bulkheads, the contractors were instructed to obtain the maximum strength with the widest possible spacing of rivets.

The above indicates that the point had been reached in United States naval construction where wider spacing of rivets in oil-tight work would have to be seriously considered and investigated in order to make practicable the construction of the torpedo bulkheads and to obtain the desired results sought in their design. This was the origin of the bureau's departure from its practice for oil-tight riveting prior to 1916.

The contractors about this time began to investigate the possibilities of increasing

the spacing of rivets for oil-tight work. One of the contractors showed by calculations that the spacing of oil-tight riveting would be somewhat greater than that allowed by the bureau for water-tight work. As a result of these calculations, permission was requested to use water-tight spacing for rivets in oil-tight work where the weight of the plating is 15 pounds and over and to use 4 diameters in seams of 14-pound plating. The bureau granted this permission except in single-riveted joints where 3½ diameters was adhered to. Prior to this, at the request of the contractors, approval had been given to use 4 diameters in the seams of shell plating and inner bottom; this plating was 15 pounds or more and the seams double riveted.

The bureau's approval to use water-tight spacing for oil-tight riveting in seams and butts resulted in the following spacing:—

Single riveted throughout, 3½ diameters.

Double riveted, 14 pounds and under, 4 diameters.

Double riveted, 15 pounds and over, 4½ diameters.

Subsequently one of the contractors requested approval of 4½ diameters in plating less than 15 pounds, basing his request on experience which convinced him that well-driven rivets, spaced close enough to hold water, would also hold oil. Approval of this spacing was held in abeyance, pending comments from other building yards. However, at the suggestion of another contractor, approval was given to use 5 diameters as a general proposition in angles to plates in order to eliminate compensating liners in the way of oil-tight floors and bulkheads. Subsequently tests on the U. S. S. Tennessee have proven satisfactory with this spacing in oil-tight floor and bulkhead bounding angles connected to 25-pound plating and over.

The building yards, in commenting on the 4½ diameter spacing in plating less than 15 pounds, did not confine their remarks to this particular question but made suggestions covering various phases of oil-tight riveting, such as riveting in floor plate and bulkhead bounding bars, bulkhead stiffeners, compensating liners, etc. These suggestions were largely based on the practice at each yard under different conditions which made it very difficult for the bureau to lay down general rules to be followed by all yards. In this connection it should be remembered that the contractor is solely responsible for the oil-tightness of the structure of the ship he builds. The contractor must build a structure and guarantee its tightness under certain specified heads, and he therefore must be cautious in his suggestions to depart from practice which has been found to be satisfactory. The bureau has assumed a liberal attitude towards approving the contractor's suggestions, provided they are well founded and not too radical a departure from practice in vogue at other building yards.

Early in 1920 the bureau advised the building yards that, in consideration of the past practices and suggestions, it believed that satisfactory results would be obtained with the following spacing:—

PLATES, SEAMS AND BUTTS.

Single, 3½ diameters.
Double, under 14 pounds, 3½ diameters.
Double and treble, 14 pounds and under 20, 4 diameters.
Double and treble, 20 pounds and over, 4½ diameters.

ANGLES TO PLATES.

Angles less than 3/8 inch thick, 4½ diameters.

Angles 3/8 inch thick and over, 5 diameters.

The navy yard, New York, proposed to use 6-diameter spacing for connecting the transverse bulkheads to the torpedo bulkheads in order to avoid the necessity of compensating liners. The bureau approved this proposal, but, to verify the safety of this spacing, directed that a test tank be constructed, representative of the structure in question, and tested under pressure. Plates 26 to 30 show the scantlings, riveting and other details of this tank.

The tank was subject to a 65-foot water-pressure head and made tight. No trouble whatever was experienced in making tight the riveting or caulking of the T-bar stiffeners and transverse oil-tight bulkheads in way of 6-diameter rivet spacing.

Upon completion of the water test, the tank was subjected to a continuous twenty-four-hour endurance test with oil at a pressure head of 65 feet. No leaks developed in the caulking or riveting in way of the 6-diameter rivet spacing.

After the completion of the above test, the tank was tested to 38 pounds per square inch. This pressure was maintained for four hours, during which period no leaks developed. This excess pressure head, equivalent to about 93 feet, was applied only to determine if leaks would develop after considerable increase in pressure.

In order to determine the advisability of adopting some of the suggestions made by the various building yards and in order to clear up the doubtful points and, further, to obtain data based upon actual test to be used in connection with the revision of the bureau's "Specification for Riveting," a second test tank was constructed at the navy yard, New York, and tested. Plate 31 shows this tank in detail. The report of this test is given in full as follows:—

"Tank was divided by a bulkhead, the two sections being designated (A) and (B), the former being the section which has a seam on the 15-pound plating double strapped. The sides of this tank are constructed with four different weights of plating; namely, 12, 15, 20 and 25 pounds. The water pressure was applied to the tank, and everything remained tight until the pressure was raised above 10 pounds, when the 12-pound plating began to deflect, causing the butt caulking in the seam to leak. It was not possible to make this seam tight, and it was decided to continue the test under those conditions.

"SECTION (A).

"Water pressure was gradually applied to this section of the tank up to 10 pounds, at which pressure the tank was perfectly tight. Pressure was gradually raised. The seam in the 12-pound plating began leaking at 12 pounds and continued to leak as the pressure was raised to 30 pounds. The water supply was then cut off for one hour, during which time the pressure dropped 2½ pounds, the leakage being entirely through the butt caulking in the 12-pound plating seam; electric welding and other connections were perfectly tight during this test. The leakage amounted to ½ gallon. The only appreciable deflection was in the 12-pound side of the tank, which deflected ¾ inch and left a permanent set of ⅛ inch after pressure was removed.

"SECTION (B).

"The pressure on this section of the tank was applied similar to the method used on section (A), the result being about the same except that the butt caulking in the seam of

the 15-pound plating leaked. The only leakage was through the butt caulking of the single-strapped seams and amounted to 1½ gallons for the hour. Deflection the same as on section (A).

"AIR TEST ON SECTION (B).

"Pressure was gradually raised up to 30 pounds. The tank was tight up to 13 pounds when the seams in the 12-pound and 15-pound plating began leaking and continued to leak during the two hours the pressure was on the tank. Pressure dropped 18 pounds in two hours. All of the leakage was through the butt caulking in the 12-pound and 15-pound seams. Electric welding, riveting and other connections were perfectly tight during this test. The 12-pound plating deflected ⅞ inch under the 30-pound pressure and retained a permanent set of ¼ inch.

"SECTION (A) AIR TEST.

"The air test on this section was similar to that on section (B), with slightly better results due to the seam of the 15-pound plating being double strapped. The pressure dropped 8¾ pounds in two hours, all of the leakage being in the butt caulking of the seam in the 12-pound plating. The electric welding, riveting and other connections were perfectly tight. Deflection and permanent set were same as on section (B).

"The method used to locate air leaks during the air tests was by painting all seams, laps, rivet points, etc., with soapy water.

"45-FOOT HEAD OIL TEST ON SECTION (A).

"The tank was filled and a constant pressure of not less than 19 pounds and more than 20 pounds was kept on the tank for twenty-four hours with the following results:—

"During the twenty-four hours the tank leaked 1¾ gallons of oil, the greater portion of which came through the butt caulking of the 12-pound plating. A small portion, however, leaked through the caulking of the bulkhead bounding bar where same was riveted to the 12-pound plating. A very slight leakage around a very few rivets was too small to measure. Electric welding on the bounding bars did not show any sign of leakage. The deflection and permanent set were about the same as when under water and air pressure. The pressure was maintained by applying air pressure to the top of the stand pipe, which was filled with oil.

"45-FOOT HEAD OIL TEST IN SECTION (B).

"Results not as satisfactory as the test on section (A), as the butt caulking in the seam of the 12-pound plating leaked continually and it was necessary to refill the stand pipe during the test. The leakage for the 24 hours was 18¾ gallons, all of which came through the butt caulking in the way of the seam strap in the 12-pound plating. A few rivets showed signs of leaking, but this leakage was too small to measure. The deflection and permanent set were about the same as when under water pressure.

"65-FOOT HEAD OIL TEST IN SECTION (A).

"Pressure gradually applied up to 30 pounds. The deflection was slightly greater in the 12-pound plating under the 30-pound pressure than under 20-pound pressure; but the leakage through the butt caulking was slightly less. The leakage during the twenty-four-hour test was only 7 pints against 1¾ gallons under twenty-four-hour test. This was

due evidently to the seam on the inside being closed by the greater deflection. All the leakage was through the butt caulking in the seam of the 12-pound plating. A few rivets showed signs of leaking, but leakage was too small to measure. Electric welding and other connections were perfectly tight. Twelve-pound side of the tank deflected about 1 inch, the permanent set being ¼ inch.

"65-FOOT HEAD OIL TEST IN SECTION (B).

"This section of the tank was tested and pressure gradually applied similar to section (A) with very poor results. The butt caulking in the single seam of the 12-pound plating leaked continually, and it was necessary to refill the stand pipe five times during the test. The riveting and other connections on the section of the tank showed up very good with the exception of a few rivets which showed some signs of leaking, too small, however, to measure. The deflection and permanent set were about the same as deflection in section (A). In view of the fact that the seam in the 12-pound plating leaked so badly, it was not considered advisable to subject this section to the gasoline test.

"45-FOOT HEAD GASOLINE TEST ON SECTION (A).

"The tank was removed from the oil house to a vacant space where danger of fire would be minimized. The tank was filled with gasoline until the stand pipe overflowed. The pressure was gradually applied until the gauge registered 20 pounds. Seam in 12-pound plating leaked as in other tests, beginning to leak shortly after the tank was filled. Efforts were made to get an accurate account of the leakage of gasoline, but this was impossible due to the gasoline evaporating. The amount of leakage caught during the twenty-four-hour test under the 20-pound pressure was 9 pints. All the leakage was through the above-mentioned seam with the exception of a slight leak around the toe of the bounding bar where same was riveted to the 15-pound plating.

"65-FOOT HEAD GASOLINE TEST ON SECTION (B).

"After completion of the twenty-four-hour test under 20-pound pressure the pressure was gradually raised to 30 pounds. The leakage was slightly greater than under 20-pound pressure but in the same place. Total leakage during test under 30-pound pressure amounted to 3¾ gallons. There were a few rivets that showed slight signs of leakage; leakage, however, was too small to measure.

"This tank as a whole showed up well under the various tests, with the exception of seam in the 12-pound plating. To overcome this defect it was decided to renew the 12-pound plate, using a lap instead of a single strapped seam, rivet-spacing the same. The tank was again tested with water, air, oil and gasoline, using 30-pound pressure as the maximum. Both compartments of the tank were perfectly tight with water, air and oil. A slight leak developed between the bounding bar and the 15-pound plating in section (A), while it was under a 30-pound gasoline test; that is, after the 30-pound pressure had been on for about an hour, pressure then dropping 2 pounds in six hours, leakage ½ gallon.

"Gasoline test pressure of 30 pounds was applied to section (B) similar to section (A) and allowed to remain on for six hours. A bad leak developed about three hours after the 30-pound pressure was on between the inner seam strap and the 15-pound plating, the leakage entering section (A) about two feet from the division bulkhead. Both corners of the bulkhead bounding bar, where same are riveted to 15-pound plating,

leaked slightly. The pressure dropped 5 pounds in six hours, leakage 3½ gallons. Deflection and permanent set the same as on previous tests."

An interesting case of the unsuitability of single butt straps for oil-tight shell plating is in the case of the shell plating of the U. S. S. Kanawha below the upper turn of bilge. As originally built, the butts of this plating in way of the oil tanks were fitted with 30-pound single straps on the inside, quadruple riveted, the shell plating itself being 28 pounds. The rivets used were ⅞ inch in diameter, and the spacing in rows of the first three rows next to the butt varied from 3 inches to 3⅞ inches; the spacing of those in the fourth row, 5 inches. In service with the vessel loaded with oil, these butts gave considerable trouble, due to leakage. At the docking of this vessel, numerous butts, seams and rivets were caulked to overcome the leakage. After these repairs the vessel went to sea and, after encountering two gales, one of unusual severity, the same trouble was experienced, *i. e.*, opening of certain butts and the weeping of certain rivets. As the first attempt to overcome this leakage was unsuccessful, the bureau directed these butts to be double strapped. The work consisted of retaining the old inside strap intact, cutting out all of the rivets of the first three rows, reaming the holes to take 1 ⅛-inch rivets, fitting a 30-pound outside strap, treble riveted, and driving 1 ⅛-inch rivets in the enlarged holes of the ⅞-inch rivets. These new outside straps were edge caulked; also all rivet points were caulked. After these repairs were made the vessel carried three cargoes overseas, and the commanding officer reported that there was no evidence of leaks, the cargoes being delivered intact as to quantity and no shortage.

In all recent ships, such as battle cruisers, battleships 49 to 54, destroyer tenders, submarine tender, fuel ships, etc., the bureau has required all butts of the shell plating in the vicinity of oil tanks to be lapped butts where the thickness of the plating will permit; otherwise the butts to be double strapped.

In 1912 the question was raised as to the benefits, if any, obtained by painting faying surfaces. This was taken under advisement, and navy yards and superintending constructors were requested, in connection with repair work on naval vessels, to observe the condition of faying surfaces. While the reports received as a result of the observations were not unanimous, they indicated the doubtful value of painting faying surfaces, especially those where the riveting was either water-tight or oil-tight. A number of cases were observed where the faying surfaces of water-tight connections showed no signs of paint, and yet the surfaces indicated no evidence of corrosion.

In 1917, at the conference held in connection with the revision of the General Specifications for building ships of the U. S. Navy, at which were present representatives of the contractors, the question came up again. Strong arguments were advanced against the painting of these surfaces. One of these arguments pointed out that the painting of faying surfaces was more detrimental than useful, as it caused dirt and small particles of material to adhere to the surfaces which carried them into the joints, it being impracticable to handle plates without the surfaces coming at times in contact with the ground. About this time Lloyd's advised the bureau that they at one time had required these surfaces to be painted but had discontinued this practice many years ago.

After careful consideration of the above, the bureau revised the requirements of its General Specifications as follows:—

"Where oil tight or water tight spacing of rivets is used, the faying surfaces need not be painted. In other cases the faying surfaces shall be painted."

It is the bureau's practice to limit the use of stop waters and oil stops to the mini-

mum. In general, they are only permitted where a nonwater-tight member passes through water-tight or oil-tight members for the purpose of preventing leakage or where caulking is impossible, due to local conditions. However, oil stops are permitted where the material is 7½ pounds nominal weight or less, in seams, laps, stapling, etc., where necessary to secure oil-tightness.

Materials of stop waters used in United States naval construction are lamp wicking or canvas, saturated with a mixture of red and white lead, or the same material soaked in boiled linseed oil and then in red lead paint. Lamp wicking or canvas, saturated with a mixture of red lead and shellac, or soaked in a mixture of pine tar and shellac, is the standard oil stop for materials over 7½ pounds; for materials 7½ pounds and less the stop is made of 10-ounce canvas, soaked for half a day in clear shellac and then coated with a mixture of red lead and shellac, or 10-ounce canvas soaked in a mixture of pine tar and shellac.

An oil-stop solution, manufactured by the Navy Yard, Mare Island, and used extensively by that yard and other yards on the west coast, is made in accordance with the following formula:—

Portland cement, pounds	1.455
Lampblack, dry, pounds	.265
Petroleum spirit, gallons	.160
Soya bean oil, gallons	.840
Litharge, pounds	.0625
Total volume, gallons	1
Weight per gallon, pounds	9

Approval has been given to use felt, dipped in a solution similar to the above, for stop waters and oil stops on scout cruisers Raleigh and Detroit.

The CHAIRMAN:—Is there any further discussion? I will now call upon Mr. Frear, the author of the paper, for any reply he may wish to make.

MR. HUGO P. FREAR:—In submitting this paper the main object was to invite discussion rather than to push innovations on the poor shipbuilder. The ground has been covered so thoroughly in the resulting discussion that I doubt if there is much to be called for in the nature of a reply beyond expressing my indebtedness to the gentlemen who have responded so generously with their valued contributions.

Captain Robert Stocker's contribution dealing exhaustively with the activities of the Bureau of Construction and Repair in connection with oil-tight work is of unusual interest and constitutes a most valuable supplement to the proceedings; the more so because it is an authoritative statement of facts which could not be gathered as accurately and completely from any other source. In oil-tight work, butts fitted with single straps have rarely given continuous satisfaction and eventually reach a point where caulking becomes ineffective. These butts should preferably be lapped or, where greater strength is required, double strapped.

The conclusion arrived at in regard to not painting faying surfaces of oil and water-tight work fully agree with the author's views; indeed, it is believed that this practice is often carried too far in the specifications of some owners in the case of nonwater-tight work. Where seams are painted the material sometimes disintegrates and not only results in reduced friction of the joint but interferes with the natural rusting up process.

Commander Van Keuren has greatly enhanced the value of the paper by further description and analysis of the tests conducted at the New York Navy Yard, together with a statement in great detail regarding present practice for oil-tight riveting for United States naval vessels at that yard.

Mr. J. W. Stewart gives many valuable suggestions and speaks with authority regarding the reasons for much of the present-day practice. I am sure his remarks will be carefully studied and found most instructive by all interested in oil-tight riveting. It is gratifying to note that Mr. Stewart agrees with so many of the views expressed in the paper.

Messrs. Rigg, Norton, and Morrell have thrown interesting side lights on practically all of the points presented. Mr. Rigg brought out a point that was in my mind in connection with double bounding bars. Double bars were introduced to back up the caulking before the advent of longitudinally framed tankers and when the bulkhead plating was not so well supported in this vicinity to resist deflection. It might be said now that practically all new tankers are longitudinally framed and that the panel of the bulkhead plating is so reduced by the bracket connections that deflection is reduced to a minimum, and on that account it is not considered that the caulking on a single bounding bar requires additional support. Mr. Morrell points out that one objection to double bounding bars is the void space between the heels of the two bars where oil, water or rust may collect and prove a source of contamination when carrying lighter cargoes. During the war the S. S. Richmond was chartered to carry four cargoes of gasoline to London. In order to clean out the tanks they were filled with distillate and pumped out several times, but as soon as the gasoline was loaded it became discolored and had to be pumped out and passed through the still for re-conditioning. The gasoline was again loaded into the tanks and became discolored a second time before it was discovered that the contamination was due to the residue lodged between the heels of the double bounding bars. In order to eliminate the source of contamination, this residue was expelled by drilling and tapping into the objectional space and forcing in polymeric with a putty pump. This was a very expensive procedure, and the necessity for it would not have arisen had a single angle or T bounding bar been fitted.

Oil-tight riveting is such an illusive subject that apparently conclusive arguments can be advanced on both sides of many of the questions that vex the shipbuilder. One operator, in explaining why he favored double bars and countersunk heads and points, stated to the writer that when they carried gasoline it was their custom to first fill alternate tanks and follow up on the dry side in each case with a gang of caulkers. This could be accomplished with equal facility in the case of single bounding bars by having the bars face each other in alternate tanks and filling the rough side tanks first. While Mr. Norton is inclined to favor double bounding bars he would use panhead rivets for the reasons stated, and caulk on one side only. This was the usual practice when double bars were first introduced and is in line with the arguments advanced in the paper regarding panhead rivets.

In regard to single riveting, we are forced to admit that this must be accepted in the case of many vessels converted and to be converted to burn oil, when carried in the double bottom, even in some instances where the test heads are as great as referred to by Mr. Rigg. It is difficult to reconcile Mr. Morrell's arguments in regard to carrying mixed cargoes in view of the fact that he is building a 20,000-ton deadweight tanker to carry one kind of oil only. Mr. Norton, in his interesting description of this 20,000-ton

tanker, makes a strong claim that it will be the most economical tanker afloat on account of being designed to carry but one kind of cargo. Tankers are so numerous now that it seems extravagant to design every one to carry mixed cargoes when this only happens occasionally. The probability is that mixed cargoes will be carried less frequently as the number of tankers increase.

The time has already arrived when tankers should specialize on particular trades, thereby saving the expense of cleaning out the tanks, in which event the center-line bulkhead need not be absolutely bone dry. If, however, the necessity arises, at infrequent intervals, for carrying mixed cargoes, it should not be a difficult operation to avoid carrying a different kind of oil on each side of the center line in the same transverse compartment.

THE CHAIRMAN:—Gentlemen, I am sure you wish me to express the thanks of the Society to Mr. Frear for his fine paper and the discussion which the paper elicited, which I do.

The secretary desires to make an announcement with reference to the excursion to the yard of the Federal Shipbuilding Company.

THE SECRETARY:—Tickets for the excursion to the yard of the Federal Shipbuilding Company at Newark, New Jersey, can be obtained on application to the assistant secretary in Room 604 of this building. The steamer Pittsburgh, of the Pennsylvania Railroad Line, will leave the pier at the foot of Cortlandt Street, at nine o'clock sharp on Saturday morning, and luncheon will be served on the ship. The Federal Shipbuilding Company will have a reception committee with badges to show the visitors around the plant.

THE CHAIRMAN:—The meeting will now stand adjourned until tomorrow morning.

THIRD SESSION.

FRIDAY MORNING, NOVEMBER 12, 1920.

The Acting President, Mr. Homer L. Ferguson, called the meeting to order at 10.30 o'clock.

ACTING PRESIDENT:—The first paper on the program this morning is No. 6, entitled "Comparative Tests of Bilge Keels and a Gyro-Stabilizer on a Model of the U. S. Aircraft Carrier Langley," by Commander William McEntee, of the Construction Corps, U. S. Navy, Member.

COMMANDER MCENTEE:—The paper may be of interest in showing the first published results of the experiments carried on with the wave-making apparatus in the Model Basin. This apparatus was installed by Admiral Taylor and was almost in working condition when he left the Model Basin. In all of our models of real ships we roll them in waves, with and without bilge keels, to determine quenching characteristics and their stability under rolling conditions. The Langley was formerly the Jupiter, being converted for an airplane carrier, and as the airplanes are to land on her deck and to fly from her deck, it is highly desirable to stabilize the ship and keep it from rolling, particularly when the machines are attempting to land on the deck.

We first made the experiments with and without bilge keels, in the ordinary course carried out for all naval vessels. The results are given, as you will see, on Plates 32 and 33. These do not require much comment, possibly the marked damping effect of a bilge keel may seem greater than what is ordinarily to be expected. It may be remarked, however, that on this particular vessel there is a very square and full midship section, with about, as I remember it, 30 per cent parallel middle body, which offers a very excellent form for fitting bilge keels. The corners of such a square section are the best possible place on which to put the bilge keels, because, in rolling, the velocity of water around the corner is very much greater than for vessels of easier bilge; so that, although the bilge keel is only 15 inches in depth and 180 feet in length, it has a decided influence in reducing the roll. As you will see, in a system of waves which would roll the vessel, without bilge keels, over an arc roll of approximately 27 degrees, the bilge keels reduce the roll to less than 10 degrees; in other words, the arc of roll with bilge keels is about one-third what it is without bilge keels.

Commander McEntee then read the paper.

COMMANDER MCENTEE:—I had some moving pictures taken of the experiments as they were carried on. Unfortunately the views are not labeled and have no legends on them to indicate when the stabilizer was thrown in and when it was thrown out. When you see the pictures on the screen, you will see the model rolling and will see it stop again. That is not due to the fact that it moves out of synchronism with the waves. When you see it stop, it means that the stabilizer has been thrown in, and when it starts rolling again the stabilizer has been released. The picture should have had a legend to indicate that particular fact.

The moving pictures were then shown on the screen.

COMPARATIVE TESTS OF BILGE KEELS AND A GYRO-STABILIZER ON A MODEL OF THE U. S. AIRCRAFT CARRIER LANGLEY.

By Commander William McEntee, Construction Corps, U. S. N., Member.

[Read at the twenty-eighth general meeting of the Society of Naval Architects and Marine Engineers, held in New York, November 11 and 12, 1920.]

1. In recent rolling experiments carried out at the U. S. Experimental Model Basin on a model of the Langley, the results obtained are believed to be of sufficient interest to naval architects to warrant making them the subject of a paper for this Society. Incidentally there is shown one of the purposes for which the wave-making equipment at the Model Basin is employed.

2. The experiments had for their object a comparison of the quenching powers of bilge keels and of a gyro-stabilizer so far as may be determined by model tests. As the Langley is being refitted to carry aeroplanes which are both to fly from and to land on her deck, it appears that the prevention of excessive rolling is a matter of greater importance than for other vessels. As an incident to the tests the increase of the towing resistance of the model when rolling was measured and the corresponding increase in effective horse-power required for the ship estimated with interesting results.

3. For use in the tests a small gyro-stabilizer shown on Plate 39 was obtained from the Sperry Gyroscope Co. This was an exhibition stabilizer on hand, and, as it was not designed especially to suit the model, it was larger than necessary. Also the speed of the precession motor was not well adapted to the period of the model to give most efficient results. The designed stabilizing moment of the twin gyros when running at 7,000 revolutions per minute is 57.4 pound-feet, but during the tests the speed was cut down as far as possible to about 2,300 revolutions per minute, reducing the stabilizing moment to about 20 pound-feet, which was still considerably larger than necessary to quench heavy rolling rapidly.

4. The dimensions of the Langley are:—Length, 520 feet; beam, 65 feet; draught, 16 feet; and 11,000 tons displacement. The model to 1/26th scale was first fitted with bilge keels, representing to scale, keels 15 inches deep by 180 feet in length and rolled in still water to obtain its declining angle curve. The bilge keels were then removed and another declining angle curve obtained. The results are shown on Plate 32, from which the quenching power of the bilge keels may be seen. Thus without bilge keels to reduce the angle of heel from 11 degrees to 3 degrees required about fifty-four swings as compared with eight swings with bilge keels. In making these tests a gyroscopic roll recorder was used so that, after the model had been inclined by hand and released, a continuous record was obtained showing how the rolling died out in the two cases. Typical curves are shown in Plate 33. In plotting angles of heel on Plate 32, the swing numbers correspond to successive hollows and

crests on these sinuous curves. In other words, one swing comprises the motion from port to starboard or vice versa.

5. The model was next rolled with and without bilge keels and with the gyro-stabilizer in waves. For this purpose it was held broadside to the waves by single head and stern lines made fast to the model at the water-line close to the longitudinal rolling axis. The height and period of the waves could be controlled within limits and could be so adjusted as to produce marked rolling. The results of these tests are shown on Plate 34. In the same system of waves the bilge keels reduce the arc of roll from 26.4 degrees to 9.6 degrees, or about two-thirds, while the gyro-stabilizer reduced it very quickly when cut in, from 34 degrees to 3 degrees or to about one-eleventh of the uncontrolled roll, which was somewhat greater than in the two preceding cases because of better synchronism between the waves and the natural period of the model.

6. The model was then tested in waves approaching half the period of the model, the resulting records being shown on Plate 35. With the bilge keels in place, when the wave length was shortened so that the rolling was no longer syncronous the model stopped rolling almost entirely, and no other wave period could be found that would induce rolling to any extent as the typical record on Plate 35 illustrates. When the bilge keels were removed, however, it was easy to find a period of wave in the vicinity of the half period of the model that would cause very heavy rolling. The angle rolled through was so great that it was necessary to use a shorter arm for the pencil of the gyroscopic roll recorder, thus changing the scale for angle of roll on the records and running up to at least 48 degrees as shown on Plate 35. The last record on Plate 35 shows the effect of the gyro-stabilizer. When cut in, the stabilizer almost instantly cut down the maximum roll to about 2 degrees.

7. As it was apparent from the preceding tests that the roll-quenching capacity of the gyro-stabilizer was more than ample for the model, an investigation was next made to determine its performance, without bilge keels, in higher waves, the higher waves being obtained by increasing the eccentricity of the wave maker while attempting to keep the period of the wave such as to produce maximum rolling. On Plate 36 are shown the rolling records for different waves. In the following table is given the eccentricity of the wave maker and the corresponding height, length and period of the waves for the ship.

TABLE I.

Wave-maker eccentricity, inches	Approximate dimension of wave for ship		
	Height, feet	Length, feet	Period, seconds
4	11.9	1,391	16.5
5	13.5	1,409	16.6
6	17.3	1,275	15.8
7	19.5	1,357	16.3

It will be seen on Plate 36 that the stabilizer was capable of controlling the roll with the largest wave that the wave maker could produce. In the case of the wave produced by the 6-inch eccentricity, the stabilized roll was 5 degrees as compared with 2½ degrees for the larger wave, corresponding to a 7-inch eccentricity. This was due to lack of proper adjustment of the control gyro. The control gyro was adjusted after rolling in the smaller wave and in the larger wave the roll was reduced to 2.5 degrees.

8. The estimated stabilizing moment of the gyro-stabilizer as run during the experiments was 20 pound-feet. This was the minimum stabilizing moment which could be obtained because the gyro-motors could not be run at any lower speed. To compare the estimated stabilizing moment with a known impressed moment, additional tests were made in which the stabilizer was operated against a rolling moment produced by a weight moving harmonically from side to side in the model through a distance of 8 inches. This weight was actuated by an electric motor placed in the model which was without bilge keels and was free to roll in still water. By controlling the speed of the motor it was possible to make the harmonic rolling moment agree with the natural period of the model. By varying the weight it was possible to increase or decrease the rolling moment. The result of this investigation is shown on Plate 37. From this it is seen that, with a maximum rolling moment of 31.33 pound-feet, the gyro-stabilizer reduced the roll from 37.5 degrees to 3 degrees.

It is to be noted, also, that it required several rolls before the angle was so reduced. When the rolling moment was increased to 39.33 pound-feet the roll was reduced by the gyro-stabilizer from 46 degrees to 5 degrees though taking a somewhat greater number of rolls to do so. When still further increase in the rolling moment was made to 47.33 pound-feet, the capacity of the gyro-stabilizer was clearly insufficient to cope with the same, the roll being reduced from 42 degrees to 34 degrees only. It is apparent from these data and from the action of the gyro-stabilizer when opposed to the rolling moments of waves, that the moments required to produce a given angle of roll in still water are considerably greater than the moment of synchronous waves when causing a similar roll. It is also clear that the gyro-stabilizer is capable of controlling a maximum harmonic rolling moment of at least 50 per cent greater than the stabilizing moment exerted by the gyro. Though the disturbing moment may temporarily exceed the stabilizing moment, it apparently does not act for sufficient time to cause much increment in the arc of roll. This appears to confirm certain claims advanced for the so-called "Active Gyro-Stabilizer."

9. As a final test an investigation was made of the increase of towing resistance of the model when caused to roll by an harmonic rolling moment applied to the model while being towed in smooth water. The results obtained, as reduced to effective horse-power curves for the ship, are shown on Plate 38. From these it appears that at a speed of 15 knots the effective horse-power would be increased from 3,000 to 3,300 when the ship is rolling through an arc of 25 degrees, and to 3,600 when rolling through an arc of 45 degrees, corresponding to an increase in

effective horse-power of 10 per cent and 20 per cent respectively. It is to be noted that this does not include the loss of power due to decrease in propeller efficiency for a twin-screw ship, when the propellers alternately approach the surface, if not actually coming out of the water. Also, as the method of towing did not permit free yawing of the model, the losses which occur on a ship, because of the increase in rudder resistance as it is moved from side to side in an effort to steer a straight course, were not present.

These results confirm experience at sea that loss of speed is found to occur when ships are rolling heavily. Under these circumstances it appears that the power and weight devoted to the means for stabilizing a ship are more than amply repaid by the saving effected in the power required to drive her.

DISCUSSION.

THE CHAIRMAN:—Gentlemen, you have heard a very excellent paper by Commander McEntee, and I desire to express the appreciation of the Society, not only for this paper but for the moving pictures which have accompanied it, and also for the pictures which accompanied the paper yesterday of Commander Saunders. These moving pictures form a distinct addition to the method of presenting information to the Society. Is there any discussion of this paper? If so, it is in order.

MR. W. W. SMITH, *Member:*—I would like to bring up the point of the effect of rolling on the propulsive efficiency. If anyone could throw light on that subject, it would be of interest, and especially so in connection with twin-screw and single-screw types of vessels.

CAPTAIN HENRY M. GLEASON, *Member:*—I would like to bring out one point for the purpose of discussion. In the experiments carried out the increase in power required to draw the model through the water apparently is very carefully worked out, and reference was made to the figures in the propulsive coefficient. Will someone shed light on the loss in the power which is undoubtedly due to the yawing of the vessel that is part of the rolling operation. It might be interesting if anyone had any data relative to that loss, so that the three losses might be estimated and thus we would get an idea of the combined loss of the whole.

PROFESSOR H. C. SADLER, *Member of Council:*—I had occasion to compare the average work of some of our Atlantic liners, over a year's work, with the results secured on trial. The two types of vessels investigated were the high-speed type and the intermediate-speed type. In both the results worked out approximately as follows: The decrease in speed over a year's work, with the same horse-power, as compared with the tank and trial results, was in the neighborhood of 9 to 10 per cent, and roughly, the other way, it took about 30 per cent increase in horse-power to maintain the same average sea speed, as obtained on the trial results.

The curves that Mr. McEntee has shown throw a little light on that subject, and a good deal of the loss in sea speed can be traced to the rolling of the ship when half loaded.

MR. SMITH:—In connection with Captain Gleason's remarks, it would seem to me desirable to express the propulsive efficiency under rolling conditions as the ratio to the propulsive efficiency under the ideal or model tank conditions, and thereby take into account not only the increase in resistance but also the decrease in the propulsive efficiency proper. If some curve or table could be arrived at whereby the designer could approximate the actual seagoing horse-power of a ship, it would be very useful, because, in designing and powering a ship, the speed that we desire is the seagoing speed. It is not the speed on the trial trip always, but usually it is the continuous seagoing speed under average conditions.

This topic has an important bearing on that subject, and I may say, in addition, of course, the fouling of the bottom has a big effect on the power of the ship, and in ordinary merchant ships which are dry-docked periodically (about every six months), it would be good practice to approximate the increased resistance at a time when the ship has been out of dock about three months, so as to get about an average seagoing power for the ship, and in addition to make any allowance for the increase in resistance and decrease in propulsive efficiency, due to the normal rolling and sea conditions that might be expected. If Commander McEntee can throw any further light on that phase, it will be interesting.

MR. ELMER A. SPERRY, *Member:*—A year ago, when I read a paper on this subject, we had made calculations as to what, in all probability, the rolling losses were, but they were so large that we hardly dared trust them. We had them checked over by two outside people, who arrived at almost our figures. When I read the paper I only used one-quarter of these values, not daring to assign what apparently the values were.

I determined, however, that we would actually find out what these rolling losses were, and Commander McEntee has shed a great deal of light on the problem through his model work. I think the shipping interests are indebted to him profoundly for aiding in clearing up this matter.

By way of reference to these losses, we have known for some years that a marked difference exists between the actual power demanded by the ships the year through and the model results, as referred to by Professor Sadler. We have also known that the model results, under the same efficient officer, attained at the tank where he had carefully calibrated motors in the model itself, were almost absolutely coordinate with results on the range in fair or favorable weather; a fraction of 1 per cent marked the total difference, if I remember correctly.

These data were secured in calm weather—of course the tank is very calm—and yet we have known for the last twelve years that there is a very large discrepancy between these, including the tank results, and the actual operating results over long periods on the high seas. Mr. Gibbs, of the International Merchant Marine, has also made some interesting observations on this very point.

Now, of course, with Commander McEntee's model—it is really quite a large model; I suppose it would support say 12 people, being 23 feet long—with this model we have no possibility of yawing, as it is towed down the course lashed both forward and astern. Of course there was no "helm" used.

I remember Admiral Taylor telling me years ago what an astonishingly small helm angle impeded a ship; a large retardation comes about through even a small helm angle. When you calculate it, you find it is about 17 pounds per square foot retardation, with a helm only three degrees over. When the ship is rolling heavily, the busiest man on the ship is the helmsman. The ship is trying to make a sinuous course, yawing continually, and the helmsman does everything he can to try to correct it, his helm being nearly hard over much of the time. When we consider the further loss due to bad angle of attack, owing to yawing, and losses due to increased pitch due to roll, which we are learning so much about now that we can suppress it entirely, we find there are three distinct and important sources of power loss that were not included in these tests, together with a fourth one remarked by Commander McEntee regarding the lifting of the propellers alternately as the ship rolls. This loss is less in still water, but there are at least three losses of efficiency in the way I have outlived.

The curves shown by Commander McEntee are exceedingly interesting.

When we put the first stabilizer on the Worden about ten years ago, we found that the stabilizer actually would bring the vessel with the masts up very promptly when the wave increments were estimated to be considerably beyond the theoretical stabilizing capacity of the installation. You must not think that there are any ghosts in the gyroscope, because there are not. When we calculate what the stabilizing and wave-quenching ability is, we find it always right there, because when we turn on the precession in the acceptance test of rolling the ship she will periodically start out and then gradually slow down, go lazily along, and one becomes convinced that it will not reach the point that theory says it should reach on any kind of indicator, but she finally and inevitably gets there. There seems absolutely no difference between the theory and practice in gyroscope moments.

The reason is apparent when it is considered that the gyroscope deals in stresses only and does not deal in power nor anything where you have to subtract or add for unknown quantities or find any factor such as resistance—these are taken care of in getting the wheel up to its spinning speed. Now if the wheel is spinning at any given speed and we precess at some known angular velocity, the stabilizing moments imparted to the ship are those calculated, absolutely to the 100 per cent mark, and we do not have to use any factors or make any deductions.

There is another point in connection with the moving pictures. I wonder if you noticed once or twice that the model shipped seas—you would notice that a splash would come over her side now and then, but only when she was rolling heavily or dipping over towards the direction of the wave-maker. That is the very thing which Sir William White, when over here in consultation on the practical working of the gyroscopic stabilizer, predicted. He said: "If you succeed in stabilizing a ship, that ship will stop shipping seas." We all felt somewhat doubtful about it, but you noticed, as soon as the stabilizer was thrown on and the deck became level with the full sea running, no splash ever came aboard. We found that with the Worden the decks were dry when she was stabilized, but when she was not stabilized, sometimes on the lee side of the ship, there being no necessary connection between the period of the ship and the period of the sea, the waves would come aboard quite frequently. I would like to have the slides now. One of them has to do with Commander McEntee's curves.

Plate 32 looks as though it were a very substantial argument for the use of bilge keels, where we find that these keels brought the model down to one degree total roll in forty

swings. So to bring out the facts in the case so that we can make proper comparisons, I herewith submit a substitute plate, which I have denominated Plate 40. In addition to the curves of Commander McEntee's Plate 32, I have added four other curves, three of which are in dotted or broken lines. Here we have introduced curves where the ship is released at twice the angle period. The dotted line at the extreme left shows the real contribution of the stabilizer as compared with the bilge keels, because here in two and one-half swings the roll has been extinguished completely instead of requiring forty swings as with the bilge keels, and this with the boat heeled to an angle of 21½ degrees instead of 11½ degrees. The dotted line 1A shows the contribution of the bilge keels when the model is released at 21½-degree heel, showing considerable roll after seventy swings.

But the most interesting comparison on this sheet is the one now to be referred to. If we turn to Plate 36, we find that the eccentricity of the wave-maker was gradually increased from 4-inch eccentricity to 5, 6 and 7 inches. When the wave-maker was at 6 inches the model rolled, without keels, through 48½ degrees, and at 7 inches it rolled through 43½ degrees owing to a trifling difference in synchronism in the action of the wave-maker with the natural period of the model. Now let us see what would have happened with the curve on the bottom of Plate 36 had we been depending on bilge keels instead of the stabilizer. A very excellent clue to this is found on Plate 34. Here we have the wave-maker working at only 3-inch eccentricity, giving the model 26.4 degrees roll without bilge keels; with bilge keels, 9.6 degrees roll, but under practically complete control with the gyro-stabilizer.

Let us now apply these same functions to the bottom curve of Plate 36. Here we see what the stabilizer can accomplish from a heel of 21¾ degrees, this constituting the solid line to the left, marked "Exp. No. 3." Compare this now with the upper broken line on the sheet, which represents the contribution of the bilge keels under the same conditions. Here we find that the ship would roll continuously, as it does in the upper curve of Plate 34, but through an angle of something like 16 degrees (heel of 8 degrees +) continuous roll. This emphasizes the fact that with the gyro-stabilizer we are enabled to secure vastly more complete results without bilge keels, saving all the power that represents the drag due to the keels, which iscontinuous and present in calm weather as well as rough.

In what marked contrast this stands to what the stabilizer does is shown in the full line to the left on the plate, marked "Exp. 3." In both of these cases we must remember the storm is still raging to the extent of rolling the ship between 43 and 48 degrees, and instead of the curve of a 19-degree uniform amplitude passing off the sheet at nearly 10 degrees, we see in the full line that within three swings practically it has completely stabilized the ship.

You understand that in all the model experiments with the stabilizer the keels were absent, just as we have it now on the more recent ships we are stabilizing; of course the bilge keel is, like the poor, always with you, in calm weather as well as rough weather, and such losses as it entails in service are always there.

Interest again attaches to Plate 37. You will find at the bottom of paragraph 3 and also the second line of paragraph 8 that the stabilizing moment was operated at 20 pound-feet. This was pitted against artificially impressed rolling moments described, which were very much in excess of the quenching moments. When one and one-half times, shown in the first curve, Plate 37, stabilizing was very complete, which was also true when the moments were increased to practically twice (39 as to 20), although it took the gyro longer to subdue the rolling, and only when the rolling moments

were two and one-third times the quenching moments was the gyro unable to stabilize.

We have known since the experiments on the Worden in 1912 that the gyro would quench rolling when the rolling moments were at least one-third greater than the quenching moments, but we never knew that it could still be depended upon when the rolling moments were twice as great.

Very great interest centers in the last of the paper, starting at paragraph 9, where Commander McEntee confirms our calculations of a year ago that the power required to move ships through the sea under conditions of rolling was about 1 per cent per degree of heel, added to which are the other losses referred to above.

With reference to Plate 32, during the year we have found ways to still further reduce the very small power required by the stabilizer, and that power is only on when there is a sea, and only in proportion to the sea. You will notice here in our latest stabilizer how very small the motor is as compared with the wheel, and this motor, you understand, is yet 100 per cent too large, made oversize for the purpose of quickly spinning up the wheel whenever there is a sea in prospect. This contrast in size of motor and wheel gives a very good idea of the little power required.

THE CHAIRMAN:—Is there any further discussion? If not, I will ask Commander McEntee if he desires to reply.

COMMANDER MCENTEE:—With reference to the point brought up by Mr. Smith, I understand that he desires to distinguish between the losses due to the rolling of a single-screw ship as compared with those of a twin-screw ship. Although we have made no experiments along that line, I should imagine that the drop in propulsive efficiency would be greater with the twin-screw ship than with the single-screw ship.

With regard to the general proposition of losses in the speed at sea compared with the speed obtained in the model test, my experience agrees with that stated by Professor Sadler, namely, for a slow-speed ship or for a medium-speed ship the actual sea speed over a period of six months or a year will be about 10 per cent less than that estimated from the model test.

Of course I do not understand from that that this is a criticism of the model-basin results, because the sea conditions are so variable, the condition of the ship's bottom is so changeable, that there is no very real ideal standard to compare it with except that which is absolutely normal and where the conditions are ideal, and that is in the model basin. In other words, you get the model-basin results and know they are the best you can hope to get. Your experience must indicate how much of a percentage you will allow to cover the average sea conditions.

I might remark, with regard to the query of Mr. Smith, that we made experiments in the model basin a number of years ago which showed the decided influence that fouling has on the resistance of the ship. These experiments were not absolutely conclusive, in that they indicated the upper limit, but the increase in resistance of the ship, even a month or two months after being painted, is really startling. I think Mr. Ferguson can confirm me in the case of the Pennsylvania, which had been out of dock only two months, and the builders' trial showed a considerable power increment over that required when the vessel was tried immediately after she had been painted.

So far as determining the actual increase in resistance of power required, due to yawing, rolling and all the miscellaneous items which go to make up the increase of

power in a seaway, I think something may be done along that line in the model basin, but we shall not get the real answer until we try a real ship. The Navy Department is arranging to conduct tests of the Henderson, and I hope, if we get the results I look for, to have a paper next year which will show the actual power and speed changes between a ship which is stabilized and a ship which is unstabilized at sea. This, after all, will be the real answer to the problem.

MR. SMITH:—The model-basin power, and the power that you obtain during a trial trip, under ideal conditions, is what you might term the ideal horse-power of the ship. The service horse-power under average conditions is another thing. What I am trying to get at is the ratio between the ideal horse-power and the service horse-power.

THE CHAIRMAN:—We are indebted to Commander McEntee for his excellent paper, and while not desiring to indulge in a discussion of it, I will say that the ordinary and unscientific method of obtaining sea speed of the ordinary vessel is to give it enough power to make about a knot more than the contract requirements call for as regards the speed. The owner usually insists on an excess of power, because his ships are built to operate on schedule; and the model conditions and the trial conditions are the best conditions, and not the conditions which the ships meet with in actual service, so that we show an excess of power to give the extra speed in unfavorable conditions, the speed being especially necessary to the owner under those circumstances. Many ships have had the reputation of making speeds at sea that have been very materially aided by the Gulf Stream and other natural causes.

The next paper, gentlemen, is No. 7, entitled "Surface Condensers," by Mr. Luther D. Lovekin, Member.

MR. LOVEKIN:—What led me towards making this special form of condenser was the number of experiments which I had been conducting on heat transmission. As mentioned in the paper, I made a number of tests with five-eighths-inch tubes, having tubes of different diameters surrounding the said five-eighths-inch tubes, and much to my surprise I found that, as far as I went, the larger the tube surrounding the five-eighths-inch tubes the better results I obtained. I did not care to experiment with any larger tube than the two-inch tube surrounding the five-eighths-inch tube, as I knew the results would be impracticable, and so in making the tests I confined myself to two sizes of tubes, the lower limit at one and one-half and the higher limit at two inches. I found, by testing a tube four feet long, with all conditions the same, one with the 1½-inch inside diameter and the other with a 2-inch inside diameter surrounding said ⅝-inch tube, that I obtained a temperature rise of 60° and 100° respectively.

This clearly proved to me that there was something in the proper spacing of the tubes in a condenser; in other words, I think we have all been inclined to space tubes too closely. I then designed an apparatus with the regular spacing, with about 100 tubes, made several tests, and finally sent it to Annapolis. I found the difference between tubes *en masse* and a single tube was considerable. I then designed the condenser for the battleship Tennessee, and I had expected to have the results of the trial of the Tennessee for this meeting, but, unfortunately, she met with an accident and has not completed her trials.

However, we had a good opportunity to make several tests on some of our ships that were built at Hog Island. The Government ordered seventy of these condensers with 8,000 square feet, of which we have fitted up ten and tested about three. The results from these trials, as you will see on page 102, of the paper, were so satisfactory to me that I felt there was nothing more to be desired.

However, the data given there and signed by our engineer, Mr. Wilson, I took exception to, because he had it roughly 12 pounds for the General Electric turbines, and I understood that 11 pounds would be more nearly correct. But the results are so good that I would be fully satisfied with a water rate of 11 pounds instead of 12, because the heat transfer is remarkably good. One of the great features I was aiming for was to be able to take the condensate away from the condenser at as near the temperature corresponding to the vacuum as possible, and I think that the results here show that it is practically the same temperature.

I know there is going to be a discussion on the question as to how much was condensed and how much water was used for circulating purposes. However, I think we all know that if we take the temperature rise in circulating water and know the British thermal units in the steam, we can figure almost as closely as we can weigh the amount of circulating water used.

As to the amount of steam, I think the General Electric Company, which is represented here, will give me the exact water rate from tests conducted at their plant. Whether or not this will be the same as we would get on board ship, I am unable to say, because there are conditions that upset the tests on board ship as compared with those made in the plant.

I have made a table of data of the river trial and then another table of the sea trial. They differ somewhat, but the results are very gratifying in each case.

In addition to the seventy condensers which we are building for the Hog Island ships, about twenty other ships are being supplied with these condensers, and three have been in use and running now for a period of about six months, in connection with the reciprocating engines. While in the case of the reciprocating engine we are not interested in high vacuum, I was interested in obtaining as high a temperature of condensate as possible, and the engineers who ran these ships noticed a remarkable difference in the temperature of the condensate on these reciprocating-engined ships as compared with the former ships; in other words, if the outboard delivery temperature was within, say, 15° of the temperature of the exhaust, the condensate would be probably within 3 or 4°; in every case it was much higher than the temperature of the outboard delivery.

I am sorry I have not been able to run a laboratory test or a more complete test on these condensers, but everybody who has anything to do with ships knows that the shipowners want the ships delivered to them at the earliest possible moment, and apart from the battleship Tennessee, on which Admiral Griffin has kindly ordered a thorough test made of this condenser, I could not see my way clear to obtain any better data or any further or better information than that taken by various disinterested engineers on our trial trips.

I want to call particular attention to the fact that on all of these trials—and we have had a number of them—we could scarcely discover the difference in vacuum at the low-pressure turbine and the bottom of the condenser. We used eleven mercury gauges in the different parts of the condenser, as will be seen. We calibrated all of these gauges, and the loss with the condenser about 12 feet from the low-pressure rotor

was practically negligible. I think some of the engineers to whom I took occasion to show the design previously will be quite surprised at this, but it has been verified in so any cases that I am sure it is a fact that the drop is practically negligible. The heat transmission running as it does, in one of the tests, at about 750 British thermal units per degree, the logarithmic mean temperature difference is remarkably good. Furthermore, in order to show just what could be done in the way of a high vaccuum, we had two ejectors; we put one in use, and we found quite an infiltration of air throughout the cast-iron casing of both the exhaust pipe and condenser. Then we put the other into use and improved the vacuum considerably. We further improved the vacuum by applying a coating of bitumastic enamel on the outside of the condenser and exhaust pipe to such an extent that, when we got through, the results with the one air ejector, with the bitumastic enamel covering, were identical with the results of the two air ejectors, without any covering, showing that we had a tremendous infiltration of air.

In making a comparison with several of our merchant ships I have used the same surface for 4,280 horse-power as I formerly did with 2,700 horse-power, and with the same results; these results, however, being the results of the ordinary trial trip and not the results of laboratory tests. Therefore I must not be confused as having stated that the condenser is 50 or 75 per cent better than any other.

These results which we obtain on the Tennessee will really give the difference. However, I am sure, in the case of the condensers I have designed previously and those which have been designed recently on this new principle, that there can be a saving of more that 25 per cent in tube surface ordinarily specified for marine condensers.

Mr. Lovekin then read abstracts from his paper.

SURFACE CONDENSERS.

By Luther D. Lovekin, Esq., Member.

[Read at the twenty-eighth general meeting of the Society of Naval Architects and Marine Engineers, held in New York, November 11 and 12, 1920.]

Engineers as a rule are always deeply interested in the conservation of natural resources, whether it be in connection with either marine or stationary appliances, and while for a long time the stationary field seemed to lead in economies, "and justly so on account of the space available," the marine field has of late received its share of attention in various directions since the advent of the turbine.

In the earlier days, when the old reciprocating engine held full sway, it was useless to consider high vacuums as there was no economy to be had by the use of a high vacuum on account of the large volumes of steam to be cared for. In other words, any vacuum above 25 inches really proved to be a loss rather than a gain as one might expect, due to the design of the ordinary reciprocating engine being such as to make it incapable of economically expanding the steam further than referred to.

Every engineer knows the great value of expanding steam as far as possible providing it can be utilized economically, and the steam turbine has enabled us to go the limit in this respect. This has opened up a new field for surface condenser designing, and remarkable strides have been made by many engineers in this direction during the past few years.

The stationary engineers of the world, with the wonderful research laboratories at their command and the further advantages in being able to make exhaustive tests on actual installations, undoubtedly led the way for improving vacuum and also for taking care of other losses incident to high vacuum, and the marine engineer has been, and should be, content to profit by their experience in developing means for insuring high vacuum under decidedly different conditions.

The marine engineer has an unlimited supply of water at his command with no cost other than the amount expended for pumping it through the condenser, while the stationary engineer has in many cases to pay for the water he uses. This has brought about various methods for using the same water over again, but this costs money and therefore gives the marine engineer an advantage.

Other items that the marine engineer has to consider seriously are weight and space, so that what might be good engineering for stationary practice in condenser design would be impossible for marine practice. I refer more particularly to tubes above 7/8 inch in diameter being used; in fact all condenser tubes for marine work are practically confined to two sizes, namely, 5/8 inch and ¾ inch outside diameter.

Apart from the design of condensers for steam turbines, marine engineers have shown little interest in designs for surface condensers, and even at the present

time we find many engineers specifying condensers having so many square feet of surface per horse-power and a certain size of tube, with no regard whatever for the principal features of condenser design, such as velocity of water and the tube length necessary for extracting the British thermal units in the steam.

One might think that we had no knowledge of what a surface condenser really is if he could hear the arguments presented by various men who are unfamiliar with the principles of heat transmission and, believe me, there are many such men who will neither take the time to study the proposition themselves nor profit from those who have made it a special study. The author is free to admit that until the past few years he was one of these men who were satisfied to design his condensers on a basis of so many square feet of surface per horse-power and trust to luck for the rest, knowing, however, that he could establish a good precedent for following such a crude practice by referring to many so-called successful installations on other ships. How many millions of dollars have been wasted in the past by following such practices will never be known, but the time has come when we must conserve both our natural as well as our physical resources, and to do this we must lay aside all prejudice and abide by facts. It is no disgrace for one engineer to know more than another about a certain subject, so why throw up our hands in horror at a new suggestion? Why not analyse or cause to be analysed new suggestions and give encouragement to those who are willing to see or bring about new suggestions instead of saying, "The old practice is good enough for me."

Some interesting figures and designs are here presented which are the result of much thought on the problem of surface condensers. Data from the official trials of the battleship Tennessee, which is fitted with main and dynamo condensers of the author's design, embodying many new and valuable features, were also to have been included, but we must be content with the showing made by the U. S. transport Cantigny, as the trial of the battleship has been delayed.

Unfortunately we had no water measurement from the Cantigny either as to steam condensed or the amount used for circulating purposes, but to one familiar with turbine designs the amount of steam used can readily be determined with a reasonable degree of accuracy without weighing. Therefore the figure given me by our engineering department from trial data sheets will be used. (See Plate 45.)

These condensers have shown such uniformly good results as to leave little further to be desired. Condensers of this same principle have been designed and built for use in connection with reciprocating engines and the tests made prove the principle to be correct.

It might be interesting to some of the author's associates to learn the reason for his departure in condenser design instead of following the beaten path which is so easy. Therefore, a few facts in connection with his work in this line will be given.

During the years of 1907 to 1910 the author made some very interesting and instructive tests in heat transmission for various purposes. These were followed by numerous tests on the same apparatus by the Engineering Experiment Station at Annapolis, Maryland. In 1915 he made a series of tests on single tubes, 5/8-inch

outside diameter, placed inside of tubes of varying diameters. With a 5/8-inch tube, about 4 feet long, placed inside of a tube about 1½-inch inside diameter, a temperature rise of 60° F. was obtained. By placing this same 5/8-inch tube inside a tube 2-inch inside diameter, a temperature rise of 100° F. was obtained with the same steam pressure and the same conditions of test. This showed conclusively that the steam space surrounding the 5/8-inch outside diameter tube was a most important factor.

The author then ran a series of tests with varying velocities of water and, much to his amazement, obtained a heat transfer of about 2,750. An apparatus was then built having about one hundred tubes, 5/8-inch outside diameter, with the conventional tube layout, and this apparatus was tested exhaustively by the engineers at the Annapolis station in both vertical and horizontal positions, these results showing a heat transfer of about 1,000. This clearly proved the advantage of having ample steam space between the tubes, so it was decided to design a surface condenser with tubes spaced well apart at the entering rows and to gradually decrease this spacing to normal at the point where the condensate left the tubes.

The experiments on the various appliances showed that it was a serious mistake to fill up an enclosure with as many tubes as it could hold and call this a proper surface condenser design, as nothing could be gained by having the steam or condensed vapor traverse a mass of tubes after it had been condensed. Why not endeavor to produce a design whereby a maximum inlet area between the tubes could be obtained, so that with a volume of steam corresponding to a vacuum of 28½ inches we would have a velocity of about 100 feet per second or less? Instead of the conventional method of blocking up the entrance area between the tubes and thereby suffering a loss due to unnecessary resistance, which reacts on the turbine and causes an unnecessary loss in efficiency, and permitting the steam to travel through a bank of tubes after it had already been condensed and allowing the cold circulating water to cool the condensate unnecessarily "as it does in all condensers of ordinary design," it was decided to reduce the depth of the bank of tubes to about half of that ordinarily used and to permit the condensate to be drained from said bank of tubes at its maximum temperature, "which should be approximately that of the temperature corresponding to the temperature of the vacuum."

Realizing the great value of reducing the volume of air and non-condensable vapors, a liberal cooling chamber was provided of efficient form, this chamber also to condense any vapors that remained and the condensate from this chamber to fall into the common condensate space, where it would be removed at the bottom of the shell. When used for battleships or the like this air-cooling space was enlarged, thereby providing about 40 per cent of the total tube surface. This enabled the outer portion of the condenser to serve as a primary condenser and the so-called air-cooling space a secondary condenser.

Thus, when the turbines are operating at less than full load, "and particularly at steam capacities below half that of full load," the primary condenser can do all the work and retain its heat in the condensate instead of falling over a bank of

tubes equal to about one-half the diameter or depth of the condenser, for no other purpose than to reach the bottom of the condenser so it can be drained off by the pump. Draining the condensate off in stages after it has been condensed appeared to be more rational, and the test data confirm this belief.

It is a matter of common occurrence with this condenser to have the temperature of the condensate and the temperature corresponding to the vacuum within a fraction of a degree, and where the temperature of the outboard delivery is kept within 5 to 8 degrees of the temperature corresponding to the vacuum, the condensate continues to keep close to the temperature corresponding to the vacuum. If a wider range of temperatures be used, the results are very pronounced; that is, if the outboard delivery temperature is within 12 to 14 degrees of the temperature of the vacuum in the condenser, the condensate will invariably show 10 to 12 degrees higher than the temperature of the outboard delivery, thus showing that when pumping more circulating water through the condenser the temperature of the condensate still remains high. It has long been known that infiltration of air through cast-iron condenser shells is the source of great trouble and annoyance with high vacuums. In order to reduce this to a minimum we made tests without any coating on the shell, then with one coat of bitumastic enamel, and finally with two coats of enamel. The effect of the final coat was an improvement of nearly 1 inch of vacuum under the same conditions.

Particular attention is called to the air channel way in these condensers, it being formed of two plates running the entire length of the space between the tube sheets, these plates being spaced apart and having their sides perforated with small holes over the entire plate. This is done so as to prevent any air pockets being formed in any part of the condenser. Where the condensate pump and air ejectors are used, such as is shown on the proposed surface condenser for United States battle cruisers, it will be noted that the condenser head is cut away so as to clear the air-ejector connection. This makes a very simple connection and yet prevents the air pipe being flooded with water. In order to ascertain the height of the water in the condenser shell, a water gauge glass can be connected between the condensate suction pipe and the shell which will readily show the height of condensate, if any, within the condenser shell.

It will be noticed in the plan referred to that small angle bars are placed in the lower edges of all the drain plates. These angle bars are not continuous but simply consist of short pieces with clear spaces of equal length between them so as to prevent the formation of a continuous sheet of water the entire length of the condenser running into the condensate space, which might interfere somewhat with the vapor and air entering the secondary zone.

It is almost unnecessary to state that the design of a surface condenser for battleships and the like which are subjected to their maximum speeds for comparatively short intervals of time is an entirely different proposition from that of the merchant ships, where the maximum speed is run the greater part of the time. In a case of the battleships we can afford to force a greater amount of water through the tubes than we can in the case of a merchant ship, for the reason that the cost

of pumping the water for the short time that these vessels run under their maximum conditions is more than offset by the reduced size, weight and space which can be obtained by using more water.

In conclusion, I wish to emphasize the fact that had the marine engineers of the world fully understood the principles of surface condenser design during the past twenty-five years, as well as they do to-day, the surface condensers for most of our reciprocating engines in the merchant service could have been built with from 25 to 50 per cent less surface and still had the same reserve factor for dirty tube surface. This would have been a valuable conservation of both our physical and our natural resources.

DISCUSSION.

THE CHAIRMAN:—Gentlemen, paper No. 5, entitled "Surface Condensers," by Mr. Luther D. Lovekin, Member, is open for discussion.

MR. PARKER M. ROBINSON, *Visitor:*—Mr. Lovekin's condenser design undoubtedly shows a great deal of thought and is a big step in advance of the old type of marine condenser, where little thought was given to the details of design and arrangement so very necessary to produce results giving a high heat transfer and a high condensate temperature.

However, it may be well to call attention to certain points regarding the performance of the Lovekin condenser.

For instance, the test results given are based on two trials of the steamship Cantigny, the first on May 20 and the second on August 14 and 15, and judging from the heat transfer obtained on those two dates, the condenser seemed to be subject to a surprising degree of deterioration in the space of less than three months. This hardly seems possible, and before accepting such a conclusion a more careful examination of the test figures is in order.

The first test seems to be incomplete in many respects, necessitating assumptions for some of the most important items. For instance, it is assumed that the turbine is developing its full capacity of 6,000 shaft horse-power throughout the trial, although no data are given relative to the steam inlet pressure and the number of nozzles in use. Then the amount of circulating water is assumed, and the number of British thermal units in the exhaust steam (988 and 999) is apparently based on the assumption that there is only a small amount of moisture present at this point, which would mean that the turbine was very inefficient.

With these important items assumed, the heat transfer is calculated to be 645 and 759 for the two parts of the test.

Turning to the second trial of August 14 and 15, it is evident that more complete data were taken as indicated by the fact that the horse-power and total steam figures are given, not in round numbers, but with an accuracy down to the horse-power and

to the pound. Figuring the heat transfer for this trial, we find that it is in the neighborhood of 500 to 540 British thermal units.

The great discrepancy between this figure and that obtained on the first test can hardly be explained as deterioration. But, in view of the many assumptions made in the first test, it is safe to conclude that its results are misleading and that the lower heat transfer figure, as obtained in the second test, is more nearly representative of the true performance of the condenser.

However, a heat transfer of something over 500 British thermal units is commendable performance and compares with the results obtained from other well-designed condensers.

Professor Edward M. Bragg, *Member:*—Perhaps some of you will remember that yesterday I said that, judging from the lines that are produced in our shipyards, only a small number of the designers are familiar with the work that has been carried on in naval tanks, and the statement in the concluding paragraph of this paper allows me to take a whack at the marine engineer.

If the marine engineers were familiar with the literature of their profession, this lack of understanding need not have existed, because as early as 1906 Professor Weighton wrote for the Institute of Naval Architects a paper describing some valuable tests made upon an experimental condenser. This condenser was small, to be sure, having only about 100 square feet of surface, and was run in connection with a small triple-expansion engine of the marine type. The temperature of the inlet water was varied from 45° to 70°. The speed of the water through the tubes varied from about 1 foot per second up to 6 feet per second, and the output of the engine was varied so that the square feet of cooling surface per horsepower varied from 0.5 to 1.5. At that time the advantage was clearly shown of leading off the condensate to one side and preventing it from dripping down through the tubes. It was clearly shown that the efficiency of the cooling surface increased directly with the velocity of the water, and that in order to have the tubes act efficiently the velocity of the steam across them should be sufficient to remove any blankets of air that tended to form at that place.

I think that the shipyards should provide their designers with easy access to the literature of their profession, because there is really no excuse for the designers continuing on in the same old path for the last twenty-five years.

I had occasion to take those results as they are presented by Professor Weighton and to work them over, and I have been able to simplify them and use them in my classes as a basis for condenser design. You may be interested to know that the results obtained from the test of this condenser with 8,000 square feet of cooling surface checks in very closely with the results obtained from Weighton's experimental condenser with 100 square feet of surface. One quantity very convenient for tabulating results is the surface-section ratio. I think no mention is made of that in this paper. If we reduce all condensers to one pass, then the surface-section ratio is the ratio of the outside surface of one tube to the cross-sectional area of that tube for the passage of water. Of course this ratio has a large value; it varies from 1,000 up to 3,000. Ordinarily in marine work we use a ratio of about 2,000, but for land practice, where water is expensive and in some cases difficult to get, we try to save all we can in that direction by using as high a ratio as 3,000.

I see in Plates 43 and 44 that enough data are given to work out the surface-section

ratio for these condensers, and this works out to be 2,030. I have not been able to find enough data to permit that ratio to be worked out for the Cantigny, but taking the results given for surface, water, and speed, the ratio works out to about 2,030. If we could have the additional information for determining this coefficient, it would be of value.

I believe the condenser for the Cantigny is rather conservative, because there is no reason why the speed of the water through the tubes should not be as high as 6 feet per second. I think that in most of the condensers that have been designed in shipyards during the last few years the speed of water through the tubes has been more nearly one foot per second, and a great deal of unnecessary surface has been placed in the condensers for this reason alone.

I would also like to point out that in condenser design you do not gain very much by trying to play safe. Practically all surface which is added to the condenser, after you have enough to condense the steam, is added at the bottom of the condenser and is effective only for cooling down the air or for chilling the condensate. If you get the temperature record for water while passing through a condenser, you will find that where the water passes in at the bottom and out at the top there is a very slight increase in temperature during the first pass, if the condenser is over-surfaced, and a very rapid rise in temperature during the second pass through the steam at the top. When you play safe by putting in a lot of surface, all you are doing is to cool down the condensate and make the temperature of the hot well lower.

I would like to ask what quantity 17 is on Plate 46, "heat per pound steam." I find from the steam tables that when the condenser pressure is 1.14, the total heat is about 1,106 British thermal units. The heat of the liquid at 83.2° is about 51, giving 1,055 British thermal units, and I am wondering how the 988 British thermal units are obtained.

Mr. Harold F. Norton, *Member*:—Professor Bragg in his remarks yesterday appeared to indicate such a lack of familiarity with the production of lines in the case of new designs in the shipyards that it seemed scarcely necessary to make any reply, but, as he has mentioned it again today, perhaps it might be well to say that when lines are to be got out in a shipyard we do not merely pick out some draughtsman and say "Please get out this set of lines for this new ship." We have, at all the principal yards, I am sure, men who are particularly familiar with that work, and whose particular business it is to get out lines for a new ship. We would like to assure Professor Bragg that these men have access to the transactions of this Society, and the transactions of every other society, and we would also like to assure the members of the Society that we take advantage of all such papers, before getting out the lines of a new ship, and that these papers are greatly appreciated.

The above is intended as an appreciation of the work of the Society and to indicate what might be said to the professor's remarks, rather than any idea that so distinguished a member of the Society and the profession is unfamiliar with shipyard practice in producing the lines of a new vessel.

Mr. Geo. A. Orrok, *Member*, and E. B. Ricketts, *Visitor* (Communicated):—The author gives some very interesting figures on heat transfer obtained in an experimental condensing outfit at the Engineering Experiment Station at Annapolis. The data,

however, are valueless for comparative purposes because the conditions under which the experiments were made are not stated.

The heat transfer of 2,750 British thermal units given in the first paragraph on page 101, if it were accomplished under high vacuum conditions, is a very remarkable result, but if this was effected under heater conditions it is no more than what is accomplished in every-day results.

Most of the condensers on which high heat transfers have been obtained heretofore have been equipped with tubes of ¾ to 1 inch diameter, and it is difficult to determine whether the high heat transfer obtained in the tests on the condenser on the Cantigny are due to the general design of the condenser or to the fact that smaller tubes were used than is practical in land installations where the item of foreign matter in the circulating water is of large importance.

There have been no direct experiments published in which an attempt has been made to determine the relative efficiency of the different diameter tubes. Theoretically the economic length of tubes should be shorter and the heat transfer higher, say about 15 per cent for each ⅛-inch smaller diameter of tube. The matter is complicated by at least three unknown factors, and it is hoped some careful experimenter will conduct a series of experiments with air-free steam having this particular point in view.

The general design of the Lovekin condenser for the battleship Tennessee, etc., embodies no fundamentally new features. The matter of providing lanes between the tubes near the steam inlet to the condenser is one that has been in use for a great many years, the writers having used such lanes in condensers some fifteen or twenty years ago and demonstrated the fact of higher heat transfer when a liberal amount of space is made for the steam to get at the tubes. The first use of "lanes" in the tube bank dates from the late seventies.

While the lanes into the tube bank are aids to an efficient condenser, they do not cut down the steam velocities materially, as the residual velocity from the turbine buckets plus the space velocity is still of the order of 600 to 800 feet per second and the velocity of the tenuous water vapors approaching the tube surface must be of the order of 1,500 to 1,900 feet per second unless seriously slowed down by air aggregations in the immediate neighborhood of the tube surface.

It is unfortunate that the author did not measure his air leakage before and after coating with bitumastic varnish. Good practice limits the air leakage into a 20,000 square-foot condenser to 3 cubic feet of free air per minute while anything above 8 cubic feet should mean an overhaul and stopping the leaks. Of this 3 cubic feet at least one is due to air brought into the system with the feed water, the remainder being true leakage air. It is probable that the leakage into the 8,000 square-foot shell exceeded 25 cubic feet of free air per minute if the vacuum was reduced so large an amount.

In the matter of baffling so as to insure the condensate draining to the hot well without passing over any more tubes than is necessary, this is a modification of the Atwood Wheeler dry-tube condenser, a number of installations of which we have had in service for about twelve years. Practically all condenser manufacturers have used modifications of this type.

The general arrangement of tube surface is very similar to that supplied by the Westinghouse Company in their radial flow type condensers, dry air suction being taken from practically the same points. The radial flow type condenser, as manufactured by the Westinghouse Company, provides a larger open space between the bank

of tubes and the outside shell which, in our opinion, is an advantage as it allows more easy access of the steam to the lower banks of tubes.

The high heat transfer obtained in the test quoted by the author is higher than any obtained on land installations with which the writers are familiar. We would, however, hesitate to credit the condenser design with this higher heat transfer until the relation between the heat transfer on $\frac{5}{8}$-inch and 1-inch tubes has been accurately determined under similar test conditions.

Mr. W. W. Smith, *Member:*—Mr. Lovekin has presented a most interesting paper. He brought out the points that condensers should be designed more scientifically, in which I entirely agree with him. There are basic formulae available for designing condensers, and I think there would be considerable advantage in using them.

All of the condensers designed in the engineering department of the Federal Yard are designed from basic formula and not from the square feet or surface per shaft horse-power. I might also call Professor Bragg's attention to this fact, and that we also use a standard form for making these computations, so that the computer or designer cannot go astray in making the design for his condenser. However, it is permissible to design condensers on the basis of square feet of surface per shaft horse-power where the conditions are standard. If your water rate is the same as previously assumed, and if your water temperatures and other variables are the same, then it is permissible to use the figure of square feet per shaft horse-power, and it is quite as accurate as the other method, since the constant was originally determined by the scientific method.

Mr. Lovekin has called attention to the unlimited supply of circulating water. It is the general practice to supply about eighty times the condensate for turbine vessels, and about forty times for steam engine vessels. There is considerable difference in the practice between these two types of machinery, and it would seem that the quantity of water for steam engine practice might be increased to some extent.

In connection with the tests presented by Mr. Lovekin, I understand that the shaft horse-power and the quantity of steam condensed were not actually measured, but were estimated from curves of previous tests that were made. In view of this I do not think we can consider these results as entirely conclusive, since his tests were not based altogether on accurate measurements.

In the table given by Mr. Lovekin he gives the heat per pound of steam as 988 in one case, and 999 in the other case. It would appear that these results are rather high for reasonably good turbines, and there is some question as to whether the heat units in the condensate were deducted in getting these figures.

I have made a number of computations for similar turbines and found that the British thermal units per pound of steam range from 940 to 950, and in this table by deducting the British thermal unit per pound of condensate—that is, 82 degrees minus the 30 degrees, giving 52 degrees—and deducting this gives us a value of 958 and 949 which would appear to me to be more nearly correct.

Mr. Lovekin gives the coefficient K as 645 and 759, which is sometimes used. On the other hand, I think it is more convenient to use a formula for the heat transfer, in which K is equal to a coefficient C times the square root of the velocity since the velocity has such a large effect on the transmission. Converting Mr. Lovekin's figures into the coefficient C by using the square root of the velocity, we get 307 in one and 306 in the other case. Both of these values are very high, and it would be interesting in this con-

nection to know the condition of the condenser tubes at the time these tests were made, since the condition of the tubes, with respect to cleanliness, has a tremendous effect on the heat transmission.

The water velocity given in the table is 4⅓ feet per second. In this connection I think it is generally better in marine condensers with ¾-inch tubes to use a water velocity between 5 and 7 feet per second. The higher water velocity in the tubes creates a turbulence in the water which materially increases the heat transfer. In a great many marine condensers, as has been stated, the water velocity is very frequently around 3 feet per second. I do not know of any case where it is as low as 1 foot per seocnd, but even 3 feet a second is lower than can be used with advantage, in my opinion.

Another point which has been brought out is the steam velocity in the condenser—that is, the velocity of steam, especially in the top rows of tubes of the condenser. The reason for reducing this velocity is so as not to have a large pressure drop from the top to the bottom of the condenser, and I think it is good practice to limit this velocity in the upper tubes of the condenser to about 100 feet per second. That can be done in a number of ways. It can be done by providing a large space for the steam to enter tubes which are equally spaced, or it can be accomplished by larger spacing of the tubes, or you can arrange large V-shaped openings into the tube tanks, permitting the steam to go down that way. The principal thing is to limit the velocity to about 100 feet per second.

The velocity of water in the tubes necessarily determines the length of the tube, and we have found that about 12 feet length of tube for two pass condensers, giving a total length of 24 feet, generally gives very good results. When the tube length is reduced the velocity will also be reduced, unless an additional pass is added to the condenser. You can get the same results with an 8-foot tube with three passes as you do with a 12-foot tube with two passes.

The efficiency of a condenser depends a great deal on the air-pump efficiency. If the air pump extracts the air efficiently, so that there is no air blanket around the tubes, the heat transmission will be much higher than if there is an air blanket which prevents the proper transmission through the tube surface.

In all condenser tests—that is, tests where you are not testing the condenser plant as a whole, but testing the condenser alone—the air-pump capacity should be greatly in excess of what would be otherwise in order to leave no question as to the air blanketing of the tubes. In a condenser test, this would materially change the results. In this connection, attention might also be called to the fact that in a great many modern condenser plants in ships, not enough attention is paid to proper air tightness, and I do not believe the present practice in installing condensers in most shipyards is as good as it ought to be. I do not think that you can say that a condenser and a vacuum system is tight unless you fill up the system with water and put it under pressure. I do not believe it is possible to get a condenser air-tight unless you do that. The fact that small leakages have such a considerable effect in high vacuum condensers makes this feature very important.

With reference to the question of coefficient of heat transfer, this depends on the cleanliness of the tubes, both inside and outside, and on the air contained in the condenser. Oil especially is one of the things which reduces the heat transmission efficiency in marine condensers. Theoretically, turbine condensers should be perfectly clean, but actually they are not always so, since most of these condensers are used for condensing the steam

for auxiliaries. This steam contains oil, which, of course, gets on the tubes of the main condenser. There is practically no oil from the main turbine, but there is oil from the auxiliaries.

There is a great variation in the coefficient C between what we may call ideal conditions, with a perfectly clean tube, inside and outside, and without any air in the condenser, and a tube that is dirty, or partially dirty, and where the air conditions are not quite as good as they ought to be. Under the ideal conditions 300 into the square root of the velocity has been obtained in a number of cases, but under actual service conditions, about 200 is as good as should be allowed, I think, to secure conservative design conditions. It is believed that it should be even lower than that; in other words, between 170 and 160 may be taken as a conservative value. This may be lower than necessary, but I think that most designers would rather err on the safe side and also have more surface than they need rather than cut it down to too small an amount.

In this connection I cannot quite agree with Professor Bragg's figures reducing the surface so much as he has, and I think we should allow from 50 to 100 per cent below the ideal conditions to approximate the conditions which actually exist in service.

There is another point in connection with Mr. Lovekin's condenser which appears to be very good, and that is the method of air extraction. The air must be removed from the bottom of the condenser in such a way as not to leave pockets, and it must be cooled to a reasonable extent so that the volume of the air will be reduced and all of the vapor possible taken out of it, so that the volume of vapor and air handled by the air pump will be reduced as much as possible, and so that we will not have to put in an unreasonably large air-extraction pump. Air pockets in the top of the condenser also should be avoided.

The performance of condensers in service is very difficult to obtain accurately, because of the difficulty in measuring accurately the various features that we require. In the first place, the horse-power and the quantity of steam consumed by a steam turbine is difficult to measure, and it is almost impossible to measure them on merchant vessels. These two items are absolutely essential to get the proper performance of the condenser.

The measurement of vacuum is a thing which is done wrong more often than it is done right. The ordinary spring gauge is not suitable for measuring vacuum accurately, and this gauge is used in the majority of cases by marine engineers in recording the vacuum in their logs. Therefore I think that those measurements of vacuum cannot be considered as being reliable, and most of the extraordinary results reported from ships are results of vacuum measured with spring gauges and are of no value. The most accurate way to measure vacuum on board a ship is to take the temperature of the steam going into the condenser by means of an accurately calibrated thermometer, which is insulated from the exhaust pipe or condenser shell, so that the heat in the surrounding metal does not influence the temperature of the thermometer. This is a very simple way to obtain the vacuum and is accurate beyond all requirements of tests or performance.

There is some difference of opinion as to the vacua that condensers should be designed for. For turbine vessels the vacuum which can be obtained depends largely on the temperature of the inlet water, and with a very high temperature of inlet water it is not feasible to maintain very high vacua.

The following seems to be in fair accordance with common practice:—For 70-degree water, design condensing plants for a vacuum of 28¼ inches; and for 85-degree

water design the condensing plant for a vacuum of 27¾ inches. In case the temperature is taken as low as 60 degrees, which is not frequent, the condenser can be designed for a vacuum as high as 28½ inches. In all cases where vessels go into the tropics, the condenser should be designed for 85-degree water. The overlooking of this feature has caused many complaints from ship operators, due to the falling off of the vacuum when the ship went into the tropical waters having temperatures around 85 degrees.

Mr. G. L. Kothny, *Member:*—There has been quite a lively discussion on Mr. Lovekin's paper on surface condensers, and very little remains to be said. I would like to call the author's attention to an apparently typographical error in Plate 48. The title of this plate is "Main Condenser of the U. S. Battleship Tennessee." Referring now to Plates 49 and 50, which bear the same title, you will note that the figures illustrated in these two plates show an entirely different picture. Plates 49 and 50 show a wrought-iron shell condenser, while Plate 48 shows a cast-iron shell condenser. I am inclined to think that the condenser illustrated on Plate 48 is that used on the U. S. A. T. Cantigny.

I also wish to call attention to Plate 44, which shows a cross-section through the Lovekin patent condenser for Pacific type transport. It will be noted that there is a very large drain hole in the plate, separating the cold air space from the hot well. A hole of such large size as shown on Plate 44 will permit vapors from the hot condensate to be drawn into the air suction and the air pump and will put a considerable burden on the latter and partly destroy the advantages of the condenser design. Instead of providing a drain hole a syphon trap should be provided which will prevent vapors from the hot well escaping into the cold-air space.

This suggestion was made by the speaker for the condensers to be installed on the U. S. A. T. vessels built at Hog Island and to my knowledge was incorporated in those condensers.

It would also be interesting to have the author state some figures relative to the cost of building his type of condenser. It is apparent that the cost will be higher than that of the ordinary cylindrical shell with one baffle over the air suction, and it would be interesting to have figures for comparison, because the cost will be an item taken seriously into consideration by the prospective user before making a decision on the type.

Mr. D. W. R. Morgan, *Visitor:*—With reference to Mr. Lovekin's paper, I wish to call attention to the discrepancies existing in both the tests, as covered by Plates 45, 46 and 47.

In Plate 45 the vacuum at the ejector is given as 29.55 in test 1, and 29.63 in test 2, whereas the vacuum at the dry suction is given as 28.99 and 28.97 respectively. The vacuum in the steam space, however, is given as an average of 28.94 and 28.93. The above readings would denote a negligible drop through the condenser, but an exceedingly high difference in pressure from the air offtake of the condenser to the ejector. It is obvious to anyone familiar with condenser readings that it is not to be expected that there were any restrictions in the air piping, but that the readings in the condenser and those taken at the dry air suction are in error.

We therefore place great importance on the test given on Plate 47, in which it will be noted that there is from .15 to .18-inch drop in vacuum through the condenser. This drop in vacuum is exceedingly large for this size of condenser, namely, 8,000 square

feet, and is about what we would expect on a well-designed 40,000 square foot surface condenser.

The author states that the temperature of the condensate corresponds within a fraction of a degree to the vacuum temperature. The point at which the condensate temperature was taken is not specified, and it is the writer's experience that false readings can easily be obtained, caused by radiation from an outside source.

We wish to emphasize the remarks made by others in the discussion as to the inconsistency of the readings, and from this fact alone no conclusive results should be drawn.

Mr. Ricketts' discussion brought out very clearly the effect of air removal on heat transmission. It would have been very interesting had the author given the air removal capacity of the air ejector, as it would be quite possible to obtain very high heat transfer by having an air ejector of abnormal capacity.

In reply to Professor Bragg's statement that water velocity of 6 feet per minute should be used in all condensers, the water velocity that should be used is the function of the total pumping head, and the correct velocity to be used can only be determined after complete analysis of the pumping requirements.

We wish to emphasize one of Mr. W. W. Smith's remarks, namely, that great care should be exercised to eliminate air leaks as much as possible in the condenser system. I have seen actual cases where, due to air leaks, the temperature of the condensate was 40 degrees less than that of the incoming steam. This wide terminal difference was considerably reduced to a point where the condensate temperature was within 3 to 5 degrees of the incoming steam temperature and the only change in operation was the reduction of leakage.

Mr. M. L. Katzenstein, *Member* (Communicated):—Mr. Lovekin's paper is of interest at this time and will, I trust, create a general desire on the part of our members to study more in detail the design of surface condensers.

The author is correct in his statement that marine practice has not kept pace with stationary practice. That is a very mild statement. We have tried for many years, without success, to impress this upon various ship and engine builders. The air leakage is what decides the performance of the condenser as well as the type and size of the air-removal apparatus. The allowance for air leakage and information to design the condensing apparatus is seldom given to the manufacturer, and often he is not consulted.

There has been a steady and most marked improvement in the design of both the condenser and auxiliaries in stationary work since the introduction of the turbine, and in that field the Worthington Company has taken the lead in practically every branch. That company was the first to separate the condensate from the dry vapors; also to adopt the cooler and to carry on extensive and expensive experiments (lasting over years) covering the necessary air passages through the tubes to give uniformly maximum surface efficiency with minimum loss through condenser.

It certainly would be desirable, as Mr. Lovekin suggests, that the designers of marine apparatus should avail themselves of experience in stationary work. It is, however, quite evident from the designs shown of Mr. Lovekin's condenser that he has himself not beeen familiar with the development of the stationary condenser. Experience will show—as it has in stationary work—that fancy drilling of tube heads and unusual arrangements of the tube such as change in angle in different sections and with

corresponding change in pitch is unnecessary; also that better results will be obtained by leaving out all the baffles except such as are necessary to protect the cooler. The straight and uniform drilling with uniform pitch can be offset, as necessary, by scientific arrangement of passages and spacing between the tubes. All this will be developed in marine work the same as it has in stationary work, provided that advantage is taken of the stationary experience.

As regards the tables of performance given in Plates 45 and 46, I do not consider these figures of value as giving data to make comparisons with other tests as usually reported; data are not given by which various essential figures can be checked. The tables are evidently prepared more with an idea of getting a general comparison rather than engineering or scientific discussion or study.

In the first place the number of vacuum gauges on a condenser is of interest and necessary to the builders who study the design, but the average vacuum in the condenser is of little importance so far as the guarantee is concerned. The only vacuum that is really desired is the one which is not stated—that is, the vacuum at the exhaust or inlet flange of the condenser. If the turbine exhaust flange bolts direct to the condenser, that figure is given, but is somewhat different from the figures assumed in calculating the performance. However, great care has to be exercised in placing gauges in different sections of the condenser to insure getting accurate readings not influenced by eddies or velocity head, and while the condenser performance might show better by taking the average vacuum when there is a large loss through the condenser, that does not interest the buyer or user of the condenser.

In the next place, tests cannot be figured by taking the average vacuum for a long run and then correcting this figure to the corresponding temperature, which varies on a different curve entirely from the vacuum curve. The heat contents of the steam entering the condenser are stated. These look rather high to us, but there is no means for checking them. Also an average steam consumption is assumed throughout the run which cannot be possible provided a uniform quantity of circulating water is used, as the different observations show as much as 2½ degrees difference in the temperature rise in the circulating water.

There is no statement given as to the method of the plus and minus corrections shown on Plate 45, and this essentially modifies even the average vacuum taken for figuring the tests. The geometric mean is also taken for determining the performance of the condenser. Ordinarily the arithmetric mean is taken for the stationary tests with which comparisons will be made, and the arithmetric mean is probably the safest for purposes of comparison under ordinary conditions of operation. In this case the range in temperature is considerable, and the discharge water is brought very close to the temperature corresponding to the vacuum. This is not the place to discuss the relative value of the arithmetric and geometric means, both of which are based on conditions which do not exist in practice.

For example, the geometric or logarithmic mean, while correct for an ideally perfect condenser, assumes that the steam temperature and heat transfer are constant and uniform throughout the condenser and that the heat transmitted at all times is directly proportional to the temperature difference between the steam and circulating water. These conditions are not met in practice as we all know. The logarithmic mean becomes of less and less value as the temperature of the discharge water approaches that of the steam. For example, if we assume the case where the discharge circulating water is

equal to the temperature of the steam, then referring to Plate 46, No. 9, $t_s - t_0$ become zero, and the fraction under No. 10 will become infinity. If we take run No. 2 where line 13 gives the logarithmic mean temperature of 11.83, we find the arithmetric mean will be 14.15, so that the logarithmic mean is only 83½ per cent of that of the arithmetric mean, and the apparent heat transfer is proportionally higher.

I dwell upon these points because Plate 46, above analyzed, gives a wrong impression as to the performance of this condenser, whereas we know that the actual performance of this condenser, with carefully taken and corrected observations and under the same conditions, is not so good as for the condenser of the standard stationary design, as can be seen by examining tests—reported in engineering society papers—of condensing plants in many shore power stations.

THE CHAIRMAN:—Is there any further discussion? The shipbuilders are glad to know that so many of the gentlemen present have this matter of condensers on their minds, because those of us who are not marine engineers have long looked on the condenser as one of the weakest points in ship machinery construction. They have given more trouble in repair work than almost anything else. If there is no further discussion, I will ask Mr. Lovekin if he desires to make reply.

MR. LOVEKIN:—With reference to the results given in the first table, I wish to state that they were made as accurately as we could possibly make them. As I stated before, we placed eleven mercurial gauges on the condensers, all the gauges were calibrated, and we changed them around, so that the results are as accurate as our engineering staff could obtain.

With regard to my omission of the steam pressure to the power, I will be glad to supply all of this, but the horse-power was exactly as stated, and it was continuously kept up to that horse-power.

The reason for the results of the final test not being as good as the original test was that we had considerable difficulty with ferrules. During the war they had green men packing the condensers who in many cases packed them so full that no room was left for the ferrules. On the second trial ten tube ferrules gave out, and some of the tubes found their way into the condenser, so I am sure that the reason for our not obtaining the high heat transfer was on account of the excessive air mixture, together with a known quantity of salt water in the condenser. The vacuum on the second test was simply taken from one point in the condenser, while on the first test we had eleven points.

As the old saying goes, there is nothing new under the sun. I have brought out a number of innovations in my lifetime and found them a collection of old applications, one part taken from one appliance and another part from another appliance, but I have always tried to make something in the aggregate that would give good results. My condenser, regardless of what some members have stated, is different from any condenser I have ever seen or heard of.

I would be glad to be supplied with sketches of the condensers referred to, in order that the members of the Society may see exactly the difference between the Lovekin condenser and other dry tube condensers. I think they will be convinced that a marked difference exists.

When used in a battleship, the primary object I had in view was to design a condenser with a primary zone and a secondary zone, realizing that when a battleship is

in use she is 90 per cent of the time at half power, or less, and the other 10 per cent at full power, and I wanted the primary condenser to do all of the work when cruising at half power and less. No other condenser, so far as I known, has ever been designed on this principle.

In regard to Mr. Robinson's remarks I discussed this matter with Captain Bookwalter, engineering manager at our Hog Island plant, and he informed me that during this trial we were compelled to use all the auxiliary machinery on board such as the large refrigerating and electric light plants in addition to the auxiliary machinery required for mere propelling machinery. As a result of this the feed-water heater could not condense all the auxiliary exhaust and fully 20,000 pounds per hour of this exhaust were condensed in the main condenser, making the total figures about 90,000 instead of 70,000 pounds per hour.

Furthermore, during this trial we had about ten leaky condenser tubes to contend with. With these conditions we were restricted to the use of one air ejector by the trial board. Complete data of the trial of May 20 are also furnished as requested (Plate 52).

In reply to Professor Bragg's remarks I wish to state that in using a velocity of water of 4.37 feet per second I did so on account of the contract requiring 28½ inches vacuum with 60° sea water; and knowing, if these ships were to be used in other than transatlantic service, that sea water of a higher temperature would be used, I left quite a margin for the increased amount of water that would be required in order to maintain a good vacuum with the higher sea temperatures. A velocity of 6 feet per second is very reasonable for a condenser of this design; in fact we are using velocities of 8 and 10 feet per second with ⅝-inch for battleships at the present time. The 988 British thermal units are derived from pounds of water per pound of steam multiplied by temperature rise, namely, $54 \times 18.3 = 988.2$.

Referring to the remarks of Messrs. Orrok and Ricketts I am quite sure, if they will look thoroughly into the design of the Tennessee condenser, that they will find it presents some fundamentally new features, as no other condenser to my knowledge embodies the feature of primary and secondary condensing zones, in which the primary zone alone is used for condensing the steam for reduced powers and the secondary zone used for air cooling.

Battleships and the like are subjected to full-speed runs but for a short period, so why have a condenser design that condenses all the steam in a portion of the condenser and then let the condensate fall over the balance of the tubes to be further cooled by the cold circulating water.

I thank Mr. Smith for his very kind remarks and the interest shown by him in the subject.

Mr. Kothny has called attention to a typographical error which will be corrected, also to the hole in the bottom of the cold air space which was left open for drainage, although originally shown with a water seal which would have been better. Regarding the cost of these condensers I fully expect the increase in efficiency of my condenser to be more than sufficient to pay for the trifling increase in cost many times over.

In regard to Mr. Morgan's remarks I think he will have ample opportunity to prove for himself any of the doubtful points he refers to as I expect to have him on a trial about December 22 of this year on a similar ship to the Cantigny.

I am not willing to concede Mr. Katzenstein's remarks on fancy drilling of tube

plates or baffles as a finality by any means, neither do I require any pointers from him as to where to put vacuum gauges on a condenser.

Furthermore, he states that "we know the actual performance of this condenser is not as good as for the condenser of the standard stationary design." I challenge him to prove this statement.

Plate 42 shows clearly an innovation in condenser design for battleships and the like that seems to have been lost sight of altogether in the discussion of my paper. This represents truly what can be termed "zone condensation" which, despite the fact of absence of proof of its value at the present time, will, in my opinion, mark a new era in surface condensers for battleships and the like where full power is used but for a small fraction of the time.

I hardly think that any engineer will venture to say that the subdivision of the surface condenser into a primary and secondary zone will not show its importance at reduced engine speeds as this would be contrary to our natural laws. Is any engineer going to be foolish enough to try and prove the value of condensate falling on tubes through which cold water is passing?

THE CHAIRMAN:—The hour for closing the morning session has arrived. The other three papers will be read this afternoon at two o'clock. I call the attention of the members present to the excursion tomorrow to the Federal Shipbuilding Company's plant at Carnegie, New Jersey. The steamer will leave the Pennsylvania Railroad Pier, at the foot of Cortlandt Street, at nine o'clock. Captains of steamers have some advantage over the members of the Society—the members meet when they get together, but the captains leave when the bell strikes. Tickets may be obtained in the office of the Society at Room 604. A luncheon will be served on board.

The Federal Shipbuilding Company will have a reception committee, with badges, to show the visitors over the plant. I also understand, through Mr. Smith, that the visitors will have the opportunity to observe the launching of a vessel there tomorrow.

There will be a meeting of the Council of the Society immediately after adjournment in this room. We will meet again at two o'clock this afternoon.

The meeting then adjourned.

FOURTH SESSION.

FRIDAY AFTERNOON, NOVEMBER 12, 1920.

Captain Joseph H. Linnard called the meeting to order at 2.15 o'clock.

THE CHAIRMAN:—The first business before the meeting this afternoon is the reading of some additional names of those proposed for election, first, as Members, and whose election has been recommended by the Council. The secretary will read the names of those recommended for election.

The Secretary read the following names:—

Members (44).

Samuel Aitken, Superintending Engineer, Moore and McCormack Company, 5 Broadway, New York, N.Y. P. O. address, 669 Springdale Avenue, East Orange, N. J.

Arthur Aldrich, Chief Hull Draughtsman, Baltimore Dry Dock and Shipbuilding Company. P. O. address, Hamilton, Md.

Lieut. Col. P. M. Anderson, Chief of Construction Division, Inland and Coastwise Waterways Service, War Department, Washington, D. C.

Charles Lees Brand, Lieutenant Commander, Construction Corps, U. S. Navy, Outside Superintendent, Navy Yard, Puget Sound, Washington.

Sidney J. Coutts, Naval Architect, Wm. Esplen Sons, 31 James Street, Liverpool, England.

Frank B. Cox, Draughtsman, Federal Shipbuilding Company. P. O. address, 549 Riverside Drive, New York, N. Y., Apartment C-C.

Chauncey D. Davis, Head Planning Department, New York Shipbuilding Corporation, South Yard, Camden, N. J. P. O. address, The Montevista, 63rd and Oxford Streets, Philadelphia, Pa.

Arthur J. Feltault, Surveyor, U. S. Bureau of Survey, 23 Liberty Street, New York, N. Y. P. O. address, 1045 Ocean Avenue, Brooklyn, N. Y.

Louis J. Hall, President, Columbian Bronze Corporation, 522 Fifth Avenue, New York, N. Y.

Ralph T. Hanson, Commander, Construction Corps, U. S. Navy, Navy Yard, Puget Sound, Wash.

William T. Hayes, Assistant Port Engineer, United States Shipping Board. P. O. address, 324 West 71st Street, New York, N. Y.

Charles H. Hills, Marine Department, Worthington Pump & Machinery Company, 115 Broadway, New York, N. Y.

William T. Holm, Engine Department, Federal Shipbuilding Company. P. O. address, 8 Watson Avenue, East Orange, N. J.

Thomas S. Howes, Superintending Engineer, France & Canada Steamship Corporation, Lawrence Boulevard, Bayside, L. I.

Lawrence B. Jackson, Marine Department, Texas Company. P. O. address, 2258 Sedgwick Avenue, New York, N. Y.

Olav E. Jorgenson, Consulting Engineer, Worthington Pump and Machinery Company, 115 Broadway, New York, N. Y.

Takushiro Kawahigashi, Lieutenant Commander, Construction Corps, I. J. N., P. O. address, 1 Madison Avenue, New York, N. Y.

Walter Cook Keenan, Naval Architect, 204 Strand Building, New Orleans, La.

Edward A. Kingston, Engineer, Merchant Shipbuilding Corporation, Chester, Pa. P. O. address, 3809 Hamilton Avenue, Philadelphia, Pa.

Lynn H. Korndorff, Vice-President, Federal Shipbuilding Corporation, Kearney, N. J.

Miles A. Libbey, Commander, U. S. Navy, Navy Department, Washington, D. C.

Thomas M. McComb, Inspector, United States Shipping Board, Box 382, Hilton Village, Va.

William McKinley, Chief Hull Draughtsman, George G. Sharp Company. P. O. address, care of Mr. Smith, 62 Monticello Avenue, Jersey City, N. J.

David L. Moffat, Superintending Engineer, Standard Shipbuilding Corporation. P. O. address, 13 Dongan Street, West New Brighton, S. I., N. Y.

John S. O'Brien, Assistant Superintendent of Construction, U. S. L. H. S. P. O. address, 145 Audubon Avenue, New York, N. Y.

John S. Patten, Vice-President, Briggs Engineering Company, 227 24th Street, Brooklyn, N. Y. P. O. address, 404 61st Street, Brooklyn, N. Y.

William Perrott, Chief Engineer, S. S. Nunsemond, 324 W. 71st Street, New York, N.Y.

Walker F. Peterson, Engineer, Production Control, 103 Edgevale Road, Roland Park, Baltimore, Md.

Stephen M. Phillips, Managing Editor, *Shipbuilding Encyclopedia;* Associate Editor, *Marine Engineering*, Woolworth Building, New York, N. Y. P. O. address, Birch Street near Central Avenue, South Ozone Park, N. Y.

J. Benton Porter, Engineering Department, General Electric Company, 120 Broadway, New York, N. Y.

S. Innes Preston, Surveyor, Lloyd's Register, 126 Higashi Machi, Kobe, Japan.

Hakon W. Ramberg, President and General Manager, Ramberg Dry Dock & Repair Company, Inc. 936 83rd Street, Brooklyn, N. Y.

Eugene A. Riotte, President and General Manager, Standard Motor Construction Company, 180 Whiton Street, Jersey City, N. J.

William S. Roberts, Surveyor, American Bureau of Shipping. P. O. address, 922 Glen Terrace, Chester, Pa.

Joseph C. Scherneck, Foreman Ship Machinist, New York Shipbuilding Corporation, South Yard. P. O. address, Wenonah, N. J.

Louis L. Smith, Assistant to Vice-President, Westinghouse Electric and Manufacturing Company, South Philadelphia Works, Chester, Pa. P. O. address, 11 Sellers Avenue, Ridley Park, Delaware County, Pa.

Paul R. Smith, Assistant Marine Superintendent, Kerr Steamship Company, 44 Beaver Street, New York, N. Y. P. O. address, 47 Willow Street, Brooklyn, N. Y.

John L. Taylor, Engineer in Charge, Marine Work, American Bridge Company, Frick Building, Pittsburgh, Pa.

Charles Thompson, Marine Superintendent, New York and Argentine Steamship Company, 50 Broadway, New York, N. Y. P. O. address, 404 Oakland Avenue, West New Brighton, S. I.

James W. Trimmer, Chief Engineer, S. S. Vesta, Standard Oil Company of New York. P. O. address, Box 118, R.F.D. No. 2, Belmar, N. Y.

William M. Truss, Superintendent of Marine Repairs, Bethlehem Shipbuilding Corporation, Bethlehem, Pa.

Eric F. Waeber, Surveyor, Bureau Veritas, 17 State Street, New York, N. Y.

George D. Wilson, Chief Engineer, S. S. Waconta, Ward Line, Pier 13, East River, New York, N. Y. P. O. address, Hull Street, Rye, N. Y.

Euclid P. Worden, Mechanical Engineer, Submarine Boat Corporation, Newark Bay Shipyard, N. J. P. O. address, 216 Ridgewood Avenue, Glen Ridge, N. J.

Wilbur H. Young, Vice-President, College Point Boat Corporation, College Point, N. Y. P. O. address, 522 Fifth Avenue, New York, N. Y.

Associates (*16*).

Hollis F. Bennett, Superintendent Outfitting, Baltimore Dry Dock and Shipbuilding Company. P. O. address, 2931 Guilford Avenue, Baltimore, Md.

Frederic J. Bryant, President, American Chain Ladder Company, 30 East 42nd Street, New York, N. Y.

Thomas F. Byrne, Manager, Stevedoring Department, Auditore & Sons, Inc., 22 Beaver Street, New York, N. Y. P. O. address, 196 Schemerhorn Street, Brooklyn, N. Y.

Ira A. Campbell, Admiralty Lawyer, 27 William Street, New York, N. Y.

Raymond J. Cushman, Assistant Superintendent Outfitting, Baltimore Dry Dock and Shipbuilding Company. P. O. address, 2737 St. Paul Street, Baltimore, Md.

Charles S. Hallock, Superintendent Repairs, Morse Dry Dock and Repair Company, 5710 4th Avenue, Brooklyn, N. Y.

Irving W. Jackson, Marine Superintendent, Worthington Pump and Machinery Corporation. P. O. address, Box 771, Westwood, N. J.

Matthew N. Johnson, Assistant Cost Engineer, Merchant Shipbuilding Corporation, 1409 Siegel Street, Chester, Pa.

Edward L. Keenan, Cost Engineer, Merchant Shipbuilding Corporation, Chester, Pa. P. O. address, 5529 Willow Avenue, Philadelphia, Pa.

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THE CHAIRMAN:—Gentlemen, you have heard the names. All those in favor of the election of these gentlemen will say "Aye." Contrary-minded, "No." The gentlemen are elected.

The next business before the meeting is the reading of paper No. 8, entitled "Rules and Regulations for Freeboard," by Mr. David Arnott, member.

Mr. Arnott presented the paper.

RULES AND REGULATIONS FOR FREEBOARD.

By David Arnott, Esq., Member.

[Read at the twenty-eighth general meeting of the Society of Naval Architects and Marine Engineers, held in New York, November 11 and 12, 1920.]

The important position now held by the United States among the maritime nations of the world, as a result of the stupendous building program so expeditiously carried out during the last three years, has had the effect of bringing all matters relative to and affecting merchant shipping into the limelight, and of these matters the load line question is certainly not the least important.

Although valuable papers, from which the development of Freeboard Rules and Regulations can be traced, and which together form a very complete historical record of load line legislation in Great Britain, have appeared from time to time in the transactions of our kindred society, the Institution of Naval Architects, this very important question of freeboard does not seem to have received, at the annual meetings of this Society, that degree of attention which it merits, as the only paper dealing with the subject to be found in the transactions is one contributed in 1897 by Mr. Lewis Nixon, entitled "Regulations for Loading Vessels."

The Load Line Bill, which is at present before the Senate (see Appendix I), and which was introduced with a view to bringing the United States into line with the other maritime nations in the matter of compulsory load lines for merchant shipping, has awakened public interest in this question, and it was considered that a short paper giving a brief history of load line legislation in other countries and outlining the development of the various rules and regulations with regard to freeboard might prove of some general and historical value to all who are interested in the safety of life and property at sea and at least serve as a basis for discussion among the members of this Society.

The freeboard of a vessel is usually defined as the height of the side above the load water-line at the middle of her length measured from the top of the deck at side, and marks the limit of loading which is considered consistent with the maintenance of a proper factor of safety under all conditions of wind and weather. Freeboard rules and regulations are an endeavor to define this limit, and one can readily understand that the problem of fixing a safe load line for a complicated floating structure moving through and supported in a constantly varying degree along its length by a fluid which is itself in motion, does not lend itself to simple mathematical treatment.

There are, however, obvious and essential considerations which must be kept in view in determining the proper freeboard for any vessel:—

(*a*) The provision of that height of platform which will prevent any undue tendency to ship water aboard with consequent deck damage and which will in-

sure the crew getting about the deck in heavy weather with, if not comfort, at least comparative safety.

(*b*) A reserve of displacement, *i. e.*, reserve or spare buoyancy, must be provided as a margin against possible leakage and entry of water into the holds.

The freeboard, as determined from the above considerations, has been termed the geometric freeboard and is the minimum freeboard allowable having regard to the vessel's dimensions, form and type.

(*c*) The vessel must have a sufficiency of structural strength to render her safe when loaded to the minimum or geometric freeboard.

As a ship, like any other structure, must be designed for the load to be carried, freeboard and strength cannot be disassociated, and the inclusion of a definite standard of strength in any freeboard regulations is essential.

(*d*) If the vessel has a deficiency of structural strength as compared with that of the freeboard standard, the freeboard must be increased in order that the reduced load carried will render her relatively as safe as the standard vessel.

Although the present Load Line Bill appears to be the first serious attempt to provide load line legislation for this country, the desirability of placing some limit on loading seems to have been recognized for many years prior to 1917, at which time the United States Shipping Board took definite action and issued instructions that load lines be marked on all vessels building and to be built for the Emergency Fleet Corporation. Marine underwriters, as one would expect, had always endeavored to see that the loading of vessels in which they were particularly interested was kept within reasonable limits, and their surveyors had instructions to report cases where, in their opinion, vessels were overloaded to an undesirable extent. The American Bureau of Shipping's "Old Rules" emphasized the importance of adhering to general principles by which the amount of freeboard for safety might be determined, and contained a general table of freeboard allowances which varied from 1½ inches per foot of depth of hold at 8 feet to 3¾ inches per foot at a registered depth of 30 feet for full scantling vessels, with one-half of this allowance for hurricane deck vessels. In the case of vessels of the latter type in which the deck below the superstructure, or hurricane deck, was the main strength deck, the minimum freeboard proposed required to be submitted to the committee before classification was granted.

Mr. Lewis Nixon, in the paper previously referred to, while criticising certain minor features of the British freeboard tables in force at that time, gave expression to his own personal opinion regarding a compulsory load line in these words:—"It must not be assumed that the general opinion of having a definite safe freeboard for vessels is wrong, for as it stands this is a very wise law, and, if it is administered impartially and without undue interference with trade, it is of the utmost value to floating property." In the discussion following the reading of his paper it transpired that a bill to regulate freeboards, particularly for lake vessels, had been introduced into Congress some years before, but was dropped when it was pointed out that most of the channels of the Great Lakes would not allow of vessels being

loaded to their proper freeboard. The opinion was also expressed that freeboard was not so vital a question in the United States as in some other countries, owing to the limited draught of harbors and the extensive use in the coasting trade of hurricane deck vessels with sides high out of the water.

It is interesting to note that as long ago as the year 1874 the enterprising builders and shipowners of the Great Lakes district met in council at Buffalo and not only came to the conclusion that the marking of a load line on both ocean-going and lake vessels was desirable, but after mature deliberation formulated rules for freeboard and drew up a set of construction rules for wooden vessels in which a laudable endeavor was made to assess the dimensions of the various timbers entering into the structure with some regard to the strength required instead of tabulating scantlings on the basis of the vessel's tonnage, as was the practice of the time, the moulding of the floor timbers, for example, being fixed by the length of the floors and the load draught.

The freeboard rules recommended by the Buffalo convention were intended to apply only to vessels of the highest class. Figs. 2 and 2a, Plates 55 and 56, show in diagrammatic form the resulting freeboards for ocean-going, lake and spar deck vessels as compared with the freeboards recommended by the 1885 British Load Line Committee. It is evident from the curves that these rules could not have been applied economically outside of the comparatively narrow range of dimensions and type of vessel in existence at that time, and it is doubtful if spar deck vessels were ever limited to the draughts corresponding to the freeboards as derived from the rules. An example showing the method of determining the freeboards is given in Appendix IV.

The first legislation in Great Britain dealing with the loading of vessels was the Merchant Shipping Act of 1871, which contained the provision that seagoing vessels must have draught marks at the stem and stern, and required that the load draught of a vessel should be recorded before leaving port. The law was useful in that after this date reliable evidence with regard to load draught was available when investigating the losses of ships at sea. Although a Royal Commission, appointed as a result of the popular agitation led by Mr. Samuel Plimsoll, reported that the enforcement of any standard freeboard regulations would be disadvantageous if not impossible, the continued interest of the public forced the passing of the Merchant Shipping Act of 1873, which provided for the detention by the Board of Trade of overloaded and improperly laden vessels. The Merchant Shipping Act of 1875 required owners of foreign going vessels to mark a circular disc on the sides at the maximum draught, to record the amount of this freeboard before clearing at the custom house and to insert this record in the agreement with the crew; the Act of 1876 made these requirements compulsory for all British vessels, except small coasting vessels under 80 gross tons, fishing boats and yachts.

It should be noted here that, although an owner was compelled by the foregoing legislation to mark load lines on his vessels, the position of the load line, as indicated by the disc, was entirely a matter of personal opinion and could be altered from voyage to voyage. The officers of the Board of Trade had power to detain

a ship which was overloaded, but as there were no legally recognized freeboard rules from which to determine the limit of loading, one can readily imagine the endless disputes which arose between owners and officials as to what constituted an overloaded ship within the meaning of the act.

A conference which was called in 1875 to ascertain the practicability of laying down rules for the guidance of the Board of Trade surveyors was abortive as far as the direct object of the meeting was concerned, although agreement was reached on certain fundamental principles which it was considered should govern the determination of freeboards. Upon the introduction of spar and awning deck vessels, Lloyd's Register of Shipping made the marking of an approved load line on these special types a condition of classification, and in 1882 published freeboard tables capable of general application which were based on reserve buoyancy percentages. At about the same time the Board of Trade issued freeboard rules for the guidance of the surveyors, and as there was also in use a scale of freeboard allowances recommended by the Liverpool Underwriters' Registry, the necessity of endeavoring to arrive at a uniform basis for determining freeboards was urgent.

The first Load Line Committee was appointed by the Board of Trade in 1883, and, after extensive investigation, reported in 1885 "that it is now practicable to frame general rules covering freeboard which will prevent dangerous overloading without unduly interfering with trade."

All obtainable evidence as to what constituted a safe loading for vessels was collected from owners and masters, and the freeboard tables recommended by the committee embodied the results of ripe practical experience within the limits of the dimensions of vessels existing at a time when a cargo vessel of 300 feet in length was considered a big boat. These rules, which were based on Lloyd's 1882 tables, were approved by the Board of Trade, and a large number of shipowners, to their credit be it said, voluntarily took steps to have freeboards marked on their vessels in accordance with the new regulations, although not legally required to do so, since it was not until 1890 that the Merchant Shipping Act of that year provided for the compulsory marking of all, except some of the smaller class of British vessels, with freeboards as recommended by the 1885 Load Line Committee. Lloyd's Register and the British Corporation were empowered by the Board of Trade to assign freeboards, and the act itself contained a clause to the effect that due regard should be paid to the views of the assigning bodies in administering the regulations and in the making of any modifications in the freeboard tables which might be found necessary. This undoubtedly has had much to do with the fact that load line administration in Great Britain has never been oppressive, and that the freeboard rules and regulations have been modified from time to time in order to keep pace with the development of shipbuilding and the evolution of new types of vessels.

The Merchant Shipping Acts of 1894 and 1906 extended the provisions of the previous acts, the 1906 Act cancelling the exemption previously granted to coasting steamers under 80 tons gross and requiring all foreign vessels in British ports to have load lines marked in accordance with the British or other freeboard regula-

tions considered equally effective. For purposes of reference, the provisions of the British Merchant Shipping Acts of 1894 and 1906 relating to load lines have been given in Appendices II and III.

It may be worthy of mention here that under the regulations made by the Board of Trade, by virtue of the authority granted under Section 443, Paragraph 2, of the Merchant Shipping Act, a vessel must be maintained in good condition to continue to hold a freeboard certificate. Freeboard certificates, in the case of classed vessels, remain in force as long as the original class is maintained, except in the event of structural alterations being carried out to deck erections, etc., which may have affected the freeboard assigned. In the case of unclassed vessels freeboard certificates are issued for a period not exceeding four years, depending on the age and condition of the vessel. In the event of a vessel's class being cancelled or alterations made to the hull structure which affect the freeboard and where the time for which a certificate was granted has expired, the freeboard certificate must be handed in for cancellation and application made for a new certificate; otherwise the Board of Trade officers may notify the collector of customs, who has power to refuse clearance of a vessel on this account. A freeboard certificate is issued only after the markings have been verified and permanently cut in on a vessel's sides. The particulars on the certificate must be entered in the official log and the freeboard certificate framed and kept aboard the ship.

The 1885 Load Line Committee's freeboard tables, which in 1890 became the legal regulations, were based on Lloyd's 1882 Tables modified to agree with the views of the Board of Trade with regard to the value of erections, provision for the height of platform in the larger vessels, and the differences in freeboard for season of the year and trade, *i. e.*, summer, winter and winter North Atlantic. The table freeboards for flush deck steamers and sailers were based on reserve buoyancy; varying from 22 per cent in the case of a steamer 120 feet in length and of standard proportions, to 35 per cent for a steamer 408 feet long, the effect of the length corrections and coefficients of fineness given in the tables being to give the same percentage of reserve buoyancy to vessels of the same length. The awning and spar deck table freeboards were determined on a basis of the strength of the Lloyd's Rule awning and spar deck vessels of that date. Figs. 1 and 1*a*, Plates 53 and 54, show the 1885 table freeboards for steamers and sailers, respectively, in comparison with the original Lloyd's 1882 and Board of Trade freeboards, these early British freeboards having been reproduced from a similar diagram in Mr. Foster King's paper read before the Institution of Naval Architects in 1906, entitled "Notes on Freeboard."

The standard of strength laid down for use with the tables was Lloyd's 1885 Rules, in which definite scantlings were tabulated for full scantling spar deck and awning deck vessels on a basis of numerals. Owing to the improvement in distribution of materials which came into vogue about the time the first British Corporation Rules were issued in 1893, so-called spar deck vessels were being built relatively as stong as the 1885 three-deck vessel and were assigned minimum freeboards on the basis of a strength comparison. A similar situation exists today

in the modern "shelter deck with freeboard" vessel, in which the scantlings are somewhat less than full scantling classification requirements, and the granting of the minimum freeboard to this type can be justified only because of the lower standard of strength allowed by the freeboard regulations.

Except that explanatory notes were added in later editions in order to insure a uniform method of calculating freeboards on the part of the assigning bodies, no major changes were made until 1898, when the tables were extended to include vessels of 50 feet moulded depth, as shown in Fig. 3, Plate 57. Methods of determining the freeboards of special types, such as turret deck and shelter deck vessels, were added to the regulations at this time, and modifications were also made in the freeboards of vessels engaged in the winter North Atlantic trade.

As the publication in 1903 by the German Marine Association of freeboard rules which differed appreciably from the British regulations strengthened the hands of those who had been urging a further revision of the British tables, it may be desirable at this stage to draw attention to the departures made from existing British practice. Before the advent of load line legislation in Germany, a large number of German vessels had freeboards assigned by one or the other of the British assigning bodies, but in 1900 the German Marine Association, having decided that some control over the loading of vessels was desirable, issued instructions to owners to report the draught of their seagoing vessels on each voyage, together with the amount, nature and stowage of cargo carried. The results of this investigation were submitted to the Germanischer Lloyd's, and in 1903 the freeboard rules proposed by that classification society, as amended in conference with representatives of the shipping and shipbuilding interests, were finally adopted and received the approval of the German government in 1904.

The German freeboard tables were, in effect, the British tables improved and modified to suit the changes which had taken place in types and construction of vessels since 1885. The German rules contained tables of freeboards for both full scantling steamers and sailers, also for awning deck vessels, but no separate table was given for spar deck vessels. The depth used with the tables was measured to the top of the deck, whether of wood or steel, and the table freeboards were given from the top of the deck to the center of the disc, *i. e.*, summer freeboards instead of winter freeboards as in the British regulations. As considerable improvement had taken place in the distribution of material with a corresponding increase in the strength of vessels with complete superstructures, the reduction in freeboards granted by the German tables to vessels of the awning deck type was justified on that account. The new German sheer standard, which was considerably above that of the British tables, was also more in line with modern practice. Fig. 7, Plate 61, shows the comparative sheers in the case of a 400-foot vessel.

While in the British tables no correction is made for excess or deficiency of sheer in the case of awning or spar deck vessels, awning deck vessels in the German tables received an allowance for excess of sheer over the standard and were penalized for deficiency of sheer in the same way as for flush deck vessels. Detailed instructions were given for the method to be adopted in assigning freeboards on the basis

of strength, the standard of strength adopted for use with the tables being the 1903 Germanischer Lloyd's Rules. Vessels carrying wood in holds and having deck cargoes were allowed less freeboards than given by the tables provided they had sufficient stability, but no method of estimating this deduction was given. Small coasting sailers under 100 tons gross were allowed to load in summer deeper than given by the tables by a maximum allowance not exceeding the reduction allowed for loading in fresh water, and no addition was made to the freeboards either for steamers or sailers engaged in the North Atlantic trade in winter.

Owing to the reduction in freeboard granted to awning deck vessels, it was found that the new German regulations gave reductions in the freeboards of vessels with long erections as compared with the British, which were so considerable that it was evident that modifications would be necessary either to the British or German tables, or both, before the governments concerned could accept each other's regulations as being mutually effective.

In order to meet the situation which had arisen owing to the introduction of the German freeboard rules, the British Board of Trade called a conference of the assigning bodies, with the result that drastic alterations were made to the British tables, which had the effect of allowing considerable reductions in the freeboards of the majority of vessels, it being estimated that the reductions in freeboards granted represented an increased carrying capacity for British shipping of 1,000,000 tons. The freeboard tables, with the exception of the spar deck table, were modified, and the amendments are shown in Fig. 3, Plate 57, which indicates clearly the considerable reductions made in the freeboards of awning deck vessels, the fairing up of the flush deck freeboard curve from 28 feet to 42 feet moulded, and the reductions made to the freeboards of the larger sized sailing ships.

The increased draughts allowed by the new awning deck table necessitated a higher standard of strength for this type, and the old standard, *i. e.*, Lloyd's 1885 awning deck rules, was augmented by the addition of heavier topside scantlings. In addition to the table alterations the allowances for erections were modified and consideration was also given to the freeboard of vessels with scuppers and other openings in the side, and it was decided that the maximum reductions under the revised regulations should be granted only where the means of closing all such openings were satisfactory, the machinery casings above the upper deck were of sufficient height and strength, and the hatchways, hatch covers, etc., were efficient and in good condition. When these conditions were not complied with, the freeboards were to be increased, due regard being given to the vessel's trade.

In 1908, with a view to bringing the German freeboard tables into line with the revised British requirements, modifications were made to the erection allowances, and more detailed information was given for dealing with the freeboards of special types, such as shelter deck vessels with tonnage openings. The provisions contained in the British 1906 rules with regard to the protection of openings, nature of appliances required for closing openings in erection bulkheads, height and strength of machinery casings and cargo hatches, and the requirements as to ports, scuppers, etc., were all included in the 1908 German rules. The British

practice of adding to the freeboards of vessels engaged in the winter North Atlantic trade was adopted, and the special consideration previously given to wood cargo vessels and small coasting sailers was cancelled. The flush deck and awning deck tables were both extended, and increases were made to the table freeboards in the case of the larger sized sailers. In 1910 the German awning deck table was modified as shown in Fig. 4, Plate 58.

The latest British Load Line Committee was appointed in 1913 to advise the Board of Trade on the attitude to be adopted by the British representatives at a forthcoming international conference on load lines, and also to investigate and report as to whether the 1906 revised tables required further revision in the light of experience. Owing to the outbreak of the war, however, this conference was never called, and for the same reason the issue of the committee's report was delayed until the end of 1915. The decisions given by Courts of Inquiry into the loss in 1912 of two vessels whose freeboards had been reduced in 1906 caused considerable comment and probably had something to do with the terms of reference. In one case the court had found that the primary cause of the loss of the ship and lives was her insufficient freeboard, and in the other, while the loss of the ship was not attributed to lack of freeboard, the opinion was expressed that the freeboard granted under the revised regulations was dangerously small. It is not the purpose of this paper to go into detail with regard to the findings of the committee, as copies of the report (published in 1916) are easily obtainable, but anyone who has studied it will agree that the report is outstanding in respect to the importance of the recommendations made, and somewhat unique from the fact that underlying reasons on which the recommendations were based are stated in detail.

In investigating the effect of the 1916 revision of the freeboard tables, every consideration was given to the record of shipping casualties and to the experience of shipowners and masters as to the behavior of their vessels since they were marked with the new load lines, the classification societies being also consulted as to the practical working of the revised freeboards over a period of seven years and their effect upon the number and nature of actual damages sustained to classed vessels. The Load Line Committee, after reviewing the evidence in the two Courts of Inquiry mentioned above, decided that insufficient freeboard had nothing to do with the loss of either vessel and came to the conclusion that there was no foundation in fact for the allegation that the 1906 revision of the freeboard tables had caused a considerable increase in the number of losses of vessels. The opinion was expressed, however, that although the freeboards allowed by the revised tables were quite sufficient to insure the safety of vessels, the reduction in freeboard in some cases had had the effect of making the ships less comfortable, owing to the greater liability for water to come on board the weather decks.

The subcommittee which was appointed to deal with the highly technical questions involved in freeboard, having decided that the present British regulations were not adapted for an international standard, proceeded with their investigations with a view to evolving new freeboard rules and regulations suitable for submission to an international conference. The existing British rules, owing to the many

additions, modifications and amendments made since the first issue in 1885, had become complicated and somewhat difficult to apply and did not treat the different types of vessels in an equitable manner. Every endeavor was made in drawing up the new freeboard regulations to make the rules simple and easy of application, and the result of the committee's labors is embodied in their proposed rules for the assignment of load lines to merchant vessels, which are undoubtedly a considerable improvement over the present regulations.

Fig. 5, Plate 59, shows the proposed new table freeboards in comparison with the present British tables, and Fig. 6, Plate 60, a comparison with the present German tables. There are now two distinct sheer standards, parabolic curves where the sheer forward is twice the sheer aft, one for vessels with forecastles and a higher standard for flush deck vessels (Fig. 7, Plate 61). Although all types of vessels are penalized for deficiency of sheer as compared with the standard, it is not proposed to grant any allowance for excess of sheer over the standard. Block coefficients at 85 per cent of the molded depth are used with the tables instead of coefficients of fineness derived from under deck tonnage. The depth to be used with the tables is taken to the top of the deck at side, whether of wood or steel, and summer freeboards are tabulated as in the German regulations. The committee recommend that the freeboards of British ships be marked from the deck at side instead of being set down from a statutory deck line in accordance with the Merchant Shipping Acts. A more logical method of assessing the value of erections which takes account of their extent, disposition, and the nature of the closing appliances, has been adopted. Fig. 8, Plate 62, shows the present British freeboards compared with the committee's proposed freeboards in the case of a 400-foot vessel with different types of erections, varying proportions of $\frac{L}{D}$ and different mean sheers. The committee lays particular stress upon the necessity of providing adequate protection for openings, and instead of proposing to make additions to the freeboards of vessels on account of deficiencies in this respect, "Conditions of Assignment of Freeboard" are laid down with regard to the strength of machinery casings, hatches, ventilators, openings in the superstructures and in the sides, such as gangway, doors, side ports, etc., which are an amplification of the present requirements.

Probably the most important part of the committee's work was the formulating of a new standard of strength to be used in association with the new freeboard rules. A technical staff was employed to carry out the necessary research work, and under the guidance of Prof. Abell, of Liverpool University, now chief surveyor of Lloyd's Register, a very thorough analysis was made of ship structural practice as determined from the 1913 Rules of Lloyd's Register, British Corporation, Bureau Veritas and Germanischer Lloyd's, as it was considered that the classification rules, embodying, as they do, the results of years of experience, were good for a wide range of loading conditions. It was ultimately found possible to express by comparatively simple formulae the average minimum requirements of ordinary practice found from the above analysis.

The proposed standard of longitudinal strength is measured by the modulus of resistance of the midship section of the vessel computed by the usual methods. The standard $\frac{I}{Y}$ is found from the formula—

$$\frac{I}{Y}=f\times b\times d,$$

where b is the breadth of the vessel, d the extreme summer draught and f is a longitudinal strength factor tabulated in terms of the vessel's length. The $\frac{I}{Y}$ through a section in way of the midship deck openings must not be less than $.9\times f\times b\times d$, so that provision is made for a definite standard of deck compensation abreast openings. Straight line formulae are given for determining the standard minimum thickness of side plating and the corresponding frame spacing for any length of vessel, together with a method for estimating the increase in shell thickness required for variations in frame spacing over the standard. As transverse strength is at least of equal importance to longitudinal strength, it is gratifying to note that an endeavor has been made to lay down a minimum standard of strength for at least one of the transverse members of the hull structure, *i. e.*, for hold framing, and that the proposed formula for the required $\frac{I}{Y}$ of the frame section takes account in some degree of the principal factors which should enter into the determination of the frame scantlings, *i. e.*, length of unsupported frames, spacing of frames, load draught and height of deck loads.

While the proposed standard of strength is probably satisfactory enough as far as it goes, it is anything but complete, as a ship could be designed to comply with the new standard and yet prove very unsatisfactory from a structural point of view. Provided its limitations are understood, the proposed standard of strength should prove distinctly useful to naval architects, but it cannot take the place of classification society rules, as nothing is said about riveting and such important considerations as the bottom framing and the thickness of deck plating in relation to beam spacing. While the size of the frames as derived from the standard is affected by the height of the bracket floors, the size of the beam knees at the lowest tier of beams is not taken into account in determining the modulus of resistance required, and although the necessity of providing sufficient stiffness in the frame apart from the $\frac{I}{Y}$ value is emphasized, the necessary modifications to scantlings on this account are left to the assigning authorities. As the strength standards laid down are for vessels built on the ordinary transverse system, their application to longitudinally framed ships will also necessarily be left to the assigning authorities. No reference is made in the report to wooden or composite vessels, and this omission may have to be remedied in any international rules in view of the considerable amount of wooden tonnage still in existence.

The proposals of the 1913–15 Load Line Committee undoubtedly represent a

distinct advance over existing freeboard regulations. While there may be differences of opinion with regard to the theoretical basis of the strength standard, there is no question about the advantage possessed by the proposed rules over those at present in force in respect to the simplicity of arrangement, lack of ambiguity and ease in the actual working out of the freeboard and strength calculations. Since there can be no finality in freeboard rules any more than in rules for ship construction, it is also a distinct advantage that the proposed rules are in a form in which they can be readily amended if and when alterations are considered advisable. The proposed new regulations will mean that freeboards will be increased for practically all vessels where the openings in the terminal bulkheads of erections are not closed by Class 1 appliances, and considerable increase in freeboards will result in the case of flush deck vessels, especially if the sheer is less than the new standard. The rules are a step in the right direction in respect to the emphasis placed on structural strength and conditions of assignment as to protection of openings which, however, can be easily and cheaply carried out in new construction. Since the publication in 1916 of the British Load Line Committee's Report, sufficient time has elapsed to give the various freeboard authorities opportunity to compare the new proposals with existing freeboard regulations, and it is quite possible that the British authorities, in the light of their further experience, may now desire to make changes from the proposals of the Load Line Committee.

This country is not handicapped by precedent in the matter of freeboard or load line legislation, so that the work of the American Load Line Committee, of which Admiral Taylor is chairman and which is at present considering the entire subject, is of the highest importance. In this connection, although it has not been found possible within the limits of this paper to deal with other than the freeboards of ordinary sea-going freighters, it might be remarked that there are particular types of vessels which constitute a special problem for this country and which are being investigated by the several sub-committees formed for the Atlantic and Gulf Coasts, Pacific Coast and Great Lakes Districts under the direction of Professors McDermott, Durand and Sadler with a view to making recommendations to the Secretary of Commerce.

APPENDIX I.

66TH CONGRESS, 1ST SESSION.

H. R. 3621

IN THE SENATE OF THE UNITED STATES.

OCTOBER 14 (calendar day, OCTOBER 16), 1919.

Read twice and referred to the Committee on Commerce.

AN ACT

To establish load lines for certain vessels.

Be it enacted by the Senate and House of Representatives of the United States of America in Congress assembled, That this act shall apply to the following vessels:

(*a*) Cargo-carrying vessels of five hundred gross tons or over loading at any port or place within the United States or its possessions for a foreign voyage by sea.

(*b*) Cargo-carrying vessels of the United States of five hundred gross tons or over loading at any foreign port or place for a voyage by sea.

(*c*) Other vessels loading at any port or place in the United States or its possessions, whenever, in the judgment of the Secretary of Commerce, the safety of such vessels and their crews requires the establishment of a legal load line.

SEC. 2. That the Secretary of Commerce is hereby authorized and directed in respect of the vessels defined in section 1 (*a*) and (*b*) of this act and is hereby authorized in respect of the vessels defined in section 1 (*c*) of this act to establish by regulations from time to time the load water lines and marks thereof indicating the maximum depth to which such vessels may safely be loaded. Such regulations shall have the force of law.

SEC. 3. That it shall be the duty of the owner and of the master of every vessel subject to this act and to the regulations established thereunder, to cause the load line or lines so established to be permanently and conspicuously marked upon the vessel in such manner as the Secretary of Commerce shall direct, and to keep the same so marked. The Secretary of Commerce shall appoint the American Bureau of Shipping, or such other corporation or association for the survey or registry of shipping as may be selected by him, to determine whether the position and manner of marking on such vessels the load line or lines so established are in accordance with the provisions of this act and of the regulations established thereunder: *Provided, however,* That, at the request of the shipowner, the Secretary of Commerce may appoint, for the purpose aforesaid, any other corporation or association for the survey or registry of shipping which the shipowner may select and the Secretary of Commerce approve; or the Secretary of Commerce may appoint for said purpose any officer of the Government, who shall perform such services as may be directed

by the Secretary of Commerce. The Secretary of Commerce may, in his discretion, revoke any appointment made pursuant to this section. Such corporation, association, or officer shall, upon approving the position and manner of marking such load line or lines, issue a certificate in a form to be prescribed by the Secretary of Commerce, that the same is in accordance with the provisions of this act and of the regulations established thereunder, and shall deliver a copy thereof to the master of the vessel. It shall be unlawful for any vessel subject to this act and to said regulations to depart from her loading port or place without bearing such mark or marks, approved and certified by such corporation, association, or officer, and without having on board a copy of said certificate.

SEC. 4. That it shall be unlawful for any vessel subject to this act and to the regulations established thereunder to be so loaded as to submerge, in salt water, the load line or lines marked pursuant to this act and to the regulations established thereunder, applicable to her voyage; or so as to submerge under like conditions the point where such load line or lines ought to be marked pursuant to the provisions of this act and of the regulations established thereunder; or so as in any manner to violate the said regulations.

SEC. 5. That whenever the Secretary of Commerce shall certify that the laws and regulations in force in any foreign country relating to load lines are equally effective with the regulations established under this act, the Secretary of Commerce may direct, on proof that a vessel of that country has complied with such foreign laws and regulations, that such vessel and her master and owner shall be exempted from compliance with the provisions of this act, except as hereinafter provided: *Provided*, That this section shall not apply to the vessels of any foreign country which does not similarly recognize the load lines established under this act and the regulations made thereunder.

SEC. 6. That it shall be the duty of the master of every vessel subject to this act and to the regulations established thereunder and of every foreign vessel exempted pursuant to section 5 of this act, before departing from her loading port or place for a voyage by sea, to enter in the official log book of such vessel (if any) a statement of the position of the load line mark applicable to the voyage in question with reference to the actual water line at the time of departing from port, as nearly as the same can be ascertained.

SEC. 7. That if any collector of customs has reason to believe, on complaint or otherwise, that a vessel subject to this act and to the regulations established thereunder is about to proceed to sea from a port in the United States or its possessions within his district, when loaded in violation of section 4 of this act; or that any vessel exempted pursuant to section 5 of this act is about to proceed to sea from such port, when loaded in violation of the laws and regulations of her country with respect to load line, he may by written order served on the master or officer in charge of such vessel detain her provisionally for the purpose of being surveyed. The collector shall then serve on the master a written statement of the grounds of her detention and shall appoint three disinterested surveyors to examine the vessel and her loading and to report to him; whereupon the said collector may release or

may, by written order served on the master or officer in charge of such vessel, detain the vessel until she has been reloaded in whole or in part so as to conform to section 4 of this act, or, in case of a vessel exempted pursuant to section 5 of this act, so as to conform to the laws and regulations of her own country with respect to load line. If the vessel be ordered detained, the master may, within five days, appeal to the Secretary of Commerce, who may, if he desires, order a further survey and may affirm, set aside, or modify the order of the collector. Clearance shall be refused to any vessel which shall have been ordered detained.

SEC. 8. (*a*) That if the owner or master of any vessel subject to this act and to the regulations established thereunder shall permit her to depart from her loading port or place without having complied with the provisions of section 3 of this act, he shall for each offense be liable to the United States in a penalty of $500. If the owner or master of any vessel exempted pursuant to section 5 of this act shall permit her to depart from her loading port or place without having the load line or lines required by the laws and regulations of the country to which she belongs marked upon her as required by said laws and regulations, he shall for each offense be liable to the United States in a penalty of $500. The Secretary of Commerce may, in his discretion, remit or mitigate any penalty imposed under this paragraph.

(*b*) If the master of any vessel subject to this act and to the regulations established thereunder, or of any foreign vessel exempted pursuant to section 5 of this act, shall fail, before departing from her loading port or place, to enter in the official log book of such vessel (if any) the statement required by section 6 of this act, he shall for each offense be liable to the United States in a penalty of $100. The Secretary of Commerce may, in his discretion, remit or mitigate any penalty imposed under this paragraph.

(*c*) If any person shall knowingly permit or cause or attempt to cause any vessel subject to this act and to the regulations established thereunder to depart, or if, being the owner, manager, agent, or master of such vessel he shall fail to take reasonable care to prevent her from departing from her loading port or place when loaded in violation of section 4 of this act; or if any person shall knowingly permit or cause or attempt to cause a foreign vessel exempted pursuant to section 5 of this act to depart, or if, being the owner, manager, agent, or master of such vessel he shall fail to take reasonable care to prevent her from departing from her loading port or place when loaded more deeply than permitted by the laws and regulations of the country to which she belongs, he shall in respect of each offense be guilty of a misdemeanor, unless her going to sea in such a condition was, under the circumstances, reasonable and justifiable; and shall be punished by a fine not to exceed $1,000 or by imprisonment not to exceed one year, or both.

(*d*) If the master of any vessel or any other person shall knowingly permit or cause or attempt to cause any vessel to depart from any port or place in the United States or its possessions in violation of any order of detention made pursuant to section 7 of this act, he shall in respect of each offense be guilty of a misdemeanor

and shall be punished by a fine not to exceed $500 or by imprisonment not to exceed three months, or both.

(*e*) If any person shall conceal, remove, alter, deface, or obliterate, or shall suffer any person under his control to conceal, remove, alter, deface, or obliterate any mark or marks placed on a vessel pursuant to this act or to the regulations established thereunder, except in the event of lawful change of said marks, or to prevent capture by an enemy, he shall, in respect of each offense, be guilty of a misdemeanor and shall be punished by a fine not to exceed $1,000, or by imprisonment not to exceed one year, or both.

(*f*) Whenever the owner, manager, agent, or master of a vessel shall become subject to a fine or penalty by way of money payment pursuant to the provisions of this act, the vessel shall also be liable therefor, and may be seized and proceeded against in the district court of the United States in any district in which such vessel shall be found.

SEC. 9. That this act shall take effect six months after the date of its approval.

Passed the House of Representatives October 15, 1919.

Attest: WM. TYLER PAGE,
Clerk.

APPENDIX II.

Extracts from British Merchant Shipping Act 1894 Relating to Load Lines.

Section 436. (1) The Board of Trade may, in any case or class of cases in which they think it expedient to do so, direct any person appointed by them for the purpose, to record, in such manner and with such particulars as they direct, the draught of water of any seagoing ship, as shown on the scale of feet on her stem and stern post, and the extent of her clear side in feet and inches, upon her leaving any dock, wharf, port, or harbor for the purpose of proceeding to sea, and the person so appointed shall thereupon keep that record and shall forward a copy thereof to the Board of Trade.

2. That record or copy, if produced out of the custody of the Board of Trade, shall be admissible in evidence in manner provided by this act.

3. The master of every British seagoing ship shall, upon her leaving any dock, wharf, port, or harbor for the purpose of proceeding to sea, record her draught of water and the extent of her clear side in the official log book (if any), and shall produce the record of any chief officer of customs whenever required by him, and if he fails without reasonable cause to produce the record shall for each offense be liable to a fine not exceeding twenty pounds.

4. The master of a seagoing ship shall, upon the request of any person appointed to record the ship's draught of water, permit that person to enter the ship and to make such inspections and take such measurements as may be requisite for the purpose of the record; and if any master fails to do so, or impedes, or suffers anyone under his control to impede, any person so appointed in the execution of his duty, he shall for each offense be liable to a fine not exceeding five pounds.

5. In this section the expression "clear side" means the height from the water to the upper side of the plank of the deck from which the depth of hold as stated in the register is measured, and the measurement of the clear side is to be taken at the lowest part of the side.

Sec. 437. (1) Every British ship (except ships under 80 tons register employed solely in the coasting trade, ships employed solely in fishing, and pleasure yachts, and ships employed exclusively in trading or going from place to place in any river or inland water the whole or part of which is in any British possession) shall be permanently and conspicuously marked with lines (in this act called deck lines) of not less than 12 inches in length and 1 inch in breadth, painted longitudinally on each side amidships, or as near thereto as is practicable, and indicating the position of each deck which is above water.

2. The upper edge of each of the deck lines must be level with the upper side of the deck plank next the waterway at the place of marking.

3. The deck lines must be white or yellow on a dark ground, or black on a light ground.

4. In this section the expression "amidships" means the middle of the length of the load water-line as measured from the fore side of the stem to the aft side of the stern post.

SEC. 438. (1) The owner of every British ship proceeding to sea from a port in the United Kingdom (except ships under 80 tons register employed solely in the coasting trade, ships employed solely in fishing, and pleasure yachts) shall, before the time hereinafter mentioned, mark upon each of her sides, amidships within the meaning of the last preceding section, or as near thereto as is practicable, in white or yellow on a dark ground, or in black on a light ground, a circular disc 12 inches in diameter, with an horizontal line 18 inches in length drawn through its center.

2. The center of this disc shall be placed at such level as may be approved by the Board of Trade below the deck line marked under this act and specified in the certificate given thereunder, and shall indicate the maximum load line in salt water to which it shall be lawful to load the ship.

3. The position of the disc shall be fixed in accordance with the tables used at the time of the passing of this act by the Board of Trade, subject to such allowance as may be made necessary by any difference between the position of the deck line marked under this act and the position of the line from which freeboard is measured under the said tables, and subject also to such modifications, if any, of the tables and the application thereof as may be approved by the Board of Trade.

4. In approving any such modifications the Board of Trade shall have regard to any representations made to them by any corporation or association for the survey or registry of shipping for the time being appointed or approved by the Board of Trade, as hereinafter mentioned, for the purpose of approving and certifying the position of the load line.

SEC. 439. If a ship is so loaded as to submerge in salt water the center of the disc indicating the load line, the ship shall be deemed to be an unsafe ship within the meaning of the provisions hereafter contained in this part of this act, and such submersion shall be a reasonable and probable cause for the detention of the ship.

SEC. 440. (1) Where a ship proceeds on any voyage from a port in the United Kingdom for which the owner is required to enter the ship outwards, the disc indicating the load line shall be marked, before so entering her, or, if that is not practicable, as soon afterwards as may be.

2. The owner of the ship shall, upon entering her outwards, insert in the form of entry a statement in writing of the distance in feet and inches between the center of this disc and the upper edge of each of the deck lines which is above that center, and if default is made in inserting that statement, the ship may be detained.

3. The master of the ship shall enter a copy of that statement in the agreement with the crew before it is signed by any member of the crew, and a superintendent shall not proceed with the engagement of the crew until that entry is made.

4. The master of the ship shall also enter a copy of that statement in the official log book.

5. When a ship to which this section applies has been marked with a disc

indicating the load line, she shall be kept so marked until her next return to a port of discharge in the United Kingdom.

SEC. 441. (1) Where a ship employed in the coasting trade is required to be marked with the disc indicating the load line, she shall be so marked before the ship proceeds to sea from any port; and the owner shall also, once in every twelve months, immediately before the ship proceeds to sea, transmit or deliver to the chief officer of customs of the port of registry of the ship a statement in writing of the distance in feet and inches between the center of the disc and the upper edge of each of the deck lines which is above that center.

2. The owner, before the ship proceeds to sea after any renewal or alteration of the disc, shall transmit or deliver to the chief officer of customs of the port of registry of the ship notice in writing of that renewal or alteration, together with such statement in writing as before mentioned of the distance between the center of the disc and the upper edge of each of the deck lines.

3. If default is made in transmitting or delivering any notice or statement under this section, the owner shall, for each offense, be liable to a fine not exceeding one hundred pounds.

4. When a ship to which this section applies has been marked with a disc indicating the load line, she shall be kept so marked until notice is given of an alteration.

SEC. 442. (1) If (*a*) any owner or master of a British ship fails without reasonable cause to cause his ship to be marked as by this part of this act required, or to keep her so marked, or allows the ship to be so loaded as to submerge in salt water the center of the disc indicating the load line; or (*b*) any person conceals, removes, alters, defaces, or obliterates, or suffers any person under his control to conceal, remove, alter, deface, or obliterate any of the said marks, except in the event of the particulars thereby denoted being lawfully altered, or except for the purpose of escaping capture by an enemy, he shall for each offense be liable to a fine not exceeding one hundred pounds.

2. If any mark required by this part of this act is in any respect inaccurate so as to be likely to mislead, the owner of the ship shall for each offense be liable to a fine not exceeding one hundred pounds.

SEC. 443. (1) The Board of Trade shall appoint the Committee of Lloyd's Register of British and Foreign Shipping, or, at the option of the owner of the ship, any other corporation or association for the survey or registry of shipping approved by the Board of Trade, or any officer of the Board of Trade specially selected by the board for that purpose, to approve and certify on their behalf from time to time the position of any disc indicating the load line, and any alteration thereof, and may appoint fees to be taken in respect of any such approval or certificate.

2. The Board of Trade may make regulations (*a*) determining the lines or marks to be used in connection with the disc, in order to indicate the maximum load line under different circumstances and at different seasons, and declaring at this part of this act is to have effect as if any such line were drawn through the center of the disc; (*b*) as to the mode in which the disc and the lines or marks to

be used in connection therewith are to be marked or affixed on the ship, whether by painting, cutting, or otherwise; (*c*) as to the mode of application for, and form of, certificates under this section; and (*d*) requiring the entry of those certificates and other particulars as to the draught of water and freeboard of the ship, in the official log book of the ship, or other publication thereof on board the ship, and requiring the delivery of copies of those entries.

3. All such regulations shall, while in force, have effect as if enacted in this act, and if any person fails without reasonable cause to comply with any such regulation made with respect to the entry, publication or delivery of copies of certificates or other particulars as to the draught of water and freeboard of a ship, he shall for each offense be liable to a fine not exceeding one hundred pounds.

4. Where in pursuance of the regulations any such certificate is required to be delivered, a statement in writing as to the disc and deck lines of a ship need not be inserted in the form of entry or transmitted or delivered to a chief officer of customs under the provisions hereinbefore contained.

SEC. 444. Where the legislature of any British possession by any enactment provides for the fixing, marking, and certifying of load lines on ships registered in that possession, and it appears to Her Majesty the Queen that that enactment is based on the same principles as the provisions of this part of this act relating to load lines, and is equally effective for ascertaining and determining the maximum load lines to which those ships can be safely loaded in salt water, and for giving notice of the load line to persons interested, Her Majesty in Council may declare that any load line fixed and marked and any certificate given in pursuance of that enactment shall, with respect to ships so registered, have the same effect as if it had been fixed, marked or given in pursuance of this part of this act.

SEC. 445. (1) Where the Board of Trade certify that the laws and regulations for the time being in force in any foreign country and relating to overloading and improper loading are equally effective with the provisions of this act relating thereto, Her Majesty in Council may direct that on proof of a ship of that country having complied with those laws and regulations, she shall not, when in a port of the United Kingdom, be liable to detention for non-compliance with the said provisions of this act, nor shall there arise any liability to any fine or penalty which would otherwise arise for non-compliance with those provisions.

2. Provided that this section shall not apply in the case of ships of any foreign country in which it appears to Her Majesty that corresponding provisions are not extended to British ships.

APPENDIX III.

EXTRACTS FROM BRITISH MERCHANT SHIPPING ACT 1906 RELATING TO LOAD LINES.

1. Sections 437 to 443 of the principal act (which relate to load line) except subsections 3 and 4 of section 440, shall, after the appointed day, apply to all foreign ships while they are within any port in the United Kingdom, as they apply to British ships without prejudice:—

(*a*) To the power of His Majesty previously to apply those provisions to the ships of any foreign country, if the government of that country so desire, under section 734 of the principal act; and

(*b*) To any direction of His Majesty in Council given under section 445 of the principal act in the case of ships of any foreign country in which the regulations in force relating to overloading and improper loading are equally effective with the provisions of the principal act.

2. Section 462 of the principal act (which relates to the detention of foreign ships)

(1) Shall apply in the case of a ship which is unsafe by reason of the defective condition of her hull, equipment, or machinery, and accordingly that section shall be construed as if the words "by reason of the defective condition of her hull, equipments, or machinery, or" were inserted before the words "by reason of overloading or improper loading"; and

(2) Shall apply with respect to any foreign ships being at any port in the United Kingdom, whether those ships take on board any cargo at that port or not.

7. The exemption of ships under 80 tons register employed solely in the coasting trade under sections 437 and 438 of the principal act (which relate to the marking of deck lines and load lines) shall cease so far as respects steamships:—

Provided that the Board of Trade may except from the provisions of this section any class of steamships, so long as they do not carry cargo, and the provisions of this section shall not apply to any steamship belonging to any class so excepted.

8. (1) Section 440 of the principal act (which relates to the time for marking load lines) shall apply to all British foreign-going ships, and, so far as it is applied by this act of foreign ships, to all foreign foreign-going ships, whether the owner is required to enter the ship outwards or not.

(2) In the case of a ship which the owner is not required to enter outwards

(*a*) The disc indicating the load line shall be marked before clearance for the ship is demanded;

(*b*) The master shall prepare a statement similar to that required to be inserted in the form of entry under subsection (2) of the said section 440, and in the case of a British ship shall enter a copy of the statement in the agree-

ment with the crew and in the official log book, and subsections (3) and (4) of that section shall apply accordingly;

(*c*) The master shall deliver a copy of the statement to the officer of customs from whom a clearance for the ship is demanded, and a clearance shall not be granted until the statement is so delivered.

(3) Where the certificate referred to in subsection (4) of section 443 of the principal act (which relates to regulations as to load line) is required to be delivered, the provisions of this section as to the statement to be prepared by the master shall not take effect.

(4) For the purpose of providing for an alteration of marks during a voyage, subsection (5) of section 440 of the principal act shall be read as if the words "or, if the mark has been altered abroad in accordance with regulations made by the Board of Trade for the purpose, marked with the mark as so altered" were added after the words "so marked," and subsection (2) of section 443 of the principal act shall be read as if the purposes for which regulations may be made under that section included provision for the alteration of marks on ships abroad.

APPENDIX IV.

Method of Calculating Freeboard by Buffalo Convention 1874–76 Rules.

Initial freeboard $= \frac{D^2}{a}$.

Where D = Depth of side from main deck gunwhale to middle of turn of bilge.

$a = 7$ for lake vessels.

$a = 6$ for ocean-going vessels.

$a = 8$ for spar deck vessels.

Length correction $= \frac{1}{10}(L - 10D)$

Excess breadth deduction $= \sqrt{B - 2D}$

The limit of loading marked on vessels sides by a belt of width $= \sqrt{\text{Depth of side.}}$

Sheer recommended not to exceed 4 feet at stem and 2 feet at sternpost.

No distinction made between steamers and sailers, and freeboard the same for summer and winter.

Example:—

Lake Vessel.

117′×27′×9′ depth of side.

Length.	*Breadth.*	*Depth of side.*	
117	27	9	11.57
$10D = -90$	$2D = -18$	$\times 9$	Length $+2.7$
27	9	81	14.27
$\frac{27}{10} = 2.7''$	$\sqrt{9} = 3''$	$\frac{81}{7} = 11.57''$	Breadth -3
			11.27″

Width of belt $= \sqrt{9} = 3''$ Freeboard to center of belt $= 11.27''$

DISCUSSION

THE CHAIRMAN:—Gentlemen, this paper, No. 8, entitled "Rules and Regulations for Freeboard," deals with a very interesting subject, and I take it all of us are of the opinion that some load-line legislation should be adopted in this country, but it is one of those delicate questions that are involved in competitive work and rates between ships of different nations, and therefore the particular methods advocated in any case may lead to a great deal of controversy. The subject is now open for discussion.

MR. E. H. RIGG, *Member*:—I think we are all indebted to Mr. Arnott for presenting to us such a full and interesting historical survey of this intricate subject. As he points out, our transactions do not contain much on this subject. The subject of freeboard is commanding a great deal of attention at the present time, for several good reasons. Some of these reasons may be summarized as follows:

1. It is as impossible to design a ship without limiting draught as it is to design a bridge without defining the loads. In routine work this fact may be somewhat obscured by the use of rules which have been based on draught conditions, but which do not themselves keep draught constantly before the shipbuilder.

2. It has lately become very much the vogue to buy and sell cargo boats and oil tankers on a deadweight basis. Can anyone figure how to arrive at the deadweight carrying capacity of a vessel without a draught or freeboard standard?

3. We are about the only great maritime power which has no freeboard standards established by governmental authority. Our vessels trading to the ports of countries having such freeboard laws are compelled to submit to such law at such port; during the war emergency this may not have always been rigidly enforced, but we can rest assured it will be now.

4. There is a region between an unquestionably safe loading and one where serious structural straining can be expected. A reasonable overload on any good engineering structure can always be met without danger of actual collapse. This is the region where differences of opinion will be met and where the interests of owner, crew, builder, and underwriter will have to be taken care of by a judicious compromise.

There are two further aspects of the matter to be considered, the national and the international; the latter is fairly clear and we now tacitly accept the present British regulations, which have the experience of about thirty years behind them and which have been accepted as a basis by several other nations.

The British have recently prepared and published a set of revised and simplified freeboard rules which represent their views of a desirable international loading scale.

When we come to the national side of the question, we find several difficulties. Any ship which goes to sea is subjected to the dangers of the sea, but it is generally conceded that the coastwise trade enjoys advantages that deep-sea ships do not; it is operating in waters that are nearer to harbors of refuge, though it is also true that no ship that has a definite run to make should be so loaded that she is only fit to dodge along from port to port in constant touch with the Weather Bureau. The vast bulk of the coastwise trade is operating at draughts well within safe limits, and such shipping should be relieved of the unfair competition of the few who may be tempted to overload.

Vessels navigating sheltered and inland waters can safely load much deeper than vessels at sea or on large sheets of exposed waters; this, too, is recognized to such an extent that freeboard regulations do not appear to be needed for such vessels; in general, all they need is enough freeboard to protect them from the wash of passing craft and the smallest orders of waves. On account of absence of wave motion of any magnitude, their longitudinal stresses are easy; but for the ordinary rough and tumble work of any large harbor they need some reserve buoyancy.

Overloaded ships are subject to the following dangers:—Loss and injury of crew, structural straining and damage, deck and hatch damage, general bad seagoing qualities, smaller chance of surviving accident, cargo damage and loss.

There is undoubtedly a feeling in some quarters that you cannot overload a steel steamer. It is true that you can load deeply and get safely through voyage after voyage with good luck as to not meeting seas heavy enough to break the ship, but, taking the situation as a whole, you are in the long run going to be met with higher and higher insurance rates to cover deck and cargo damages. It then becomes a question of compromise between the contending claims. Suitable freeboard is part of this compromise, other points being strength of structure and details proportioned to such loading and enough power to cope with heavy weather; a vessel must have enough reserve buoyancy to ride the seas without taking undue quantities of water on deck. Just what is the proper point at which to stop loading a vessel is not an easy question to answer. Experience has taught us a great deal. We have seen also changes in rules to meet the competition and easier rules of rival nations, until now the seagoing brotherhood, which includes designers, builders, repairers, owners, operators, underwriters and crews, are in a position to draw conclusions based on both theory and practice, a combination that is very hard to beat.

As an indication of what effect increase in draught will have on stresses, a large tanker laid down for 27 feet has been taken as an example, starting at 26 feet and increasing by 2-foot increments to 32 feet. The other factors affected vary as follows:—

Actual draught	*Column A, draught*	*Column B*	*Column C*
26 feet	100.0	100.0	100.0
28 feet	107.7	107.7	112.25
30 feet	115.4	115.4	124.50
32 feet	123.1	123.1	137.15

Column B gives values for maximum bending moment; also maximum stress in topsides, with constant section modulus.

Column C gives increases in deadweight.

It is seen that stress goes up directly as draught but that deadweight increases faster. The vessel showed a designed maximum stress of 15,500 pounds per square inch compression on deck at 27 feet draught; loading to 30 feet increased this to 17,250 pounds. Even though theoretical stresses are seldom met with in actual service, there is a distinct warning that seaways calculated to find the weak spots will much oftener be met with at 30 feet draught than at 27 feet.

A question was asked during the discussion of Mr. Norton's paper yesterday as to the "why" of the straight sheer shown for that vessel. The manufacturing advantage was pointed out by the author, but the further structural advantage was not made as

clear as it might have been. Straight sheer means an increase in moulded depth amidships to get the same draught. I have not seen the figures for this particular vessel, but judge the midship depth is at least 12 inches more on this account. This helps the section modulus, and, with the amidship pump room, will take care of the somewhat marked concentration of the cargo amidships This is a freeboard point and needs no apology for being referred to in this discussion.

Gentlemen, I want to take this opportunity to thank you for the honor you have recently conferred on me by electing me to the Council and to assure you that I will serve you to the best of my ability.

MR. ALBERT SAUNDERS, *Member:*—There is one point which I would like to have Mr. Arnott cover. We all know that the danger point in ships is not always the deep loading. To what extent has consideration been given to the light load line?

MR. HUGO P. FREAR, *Member of Council:*—I have just entered the room and labor under the disadvantage of not having read the whole of this interesting paper or heard the previous discussion. The paper is principally historical, and Mr. Arnott very appropriately includes in Appendix I the Act "to establish load lines for certain vessels identified as H. R. 3621." This act, primarily to establish load lines for ocean foreign-going cargo vessels of 500 gross tons or over, readily passed the House and is now before the Senate Committee. Serious opposition, however, developed which prevented the passage of the act by the Senate at the last session. This opposition came principally from the owners of schooners, the smaller types of coasting vessels and even the owners of tugboats and vessels navigating inland waters only, which latter type in all probability would be exempt, as it was feared some of these vessels might be adversely affected.

As Mr. Arnott states, subcommittees have been formed to investigate special types of vessels in the coasting trades covering the Atlantic and Gulf coasts, Pacific coast and Great Lakes districts. The latest British load-line committee was appointed by the Board of Trade to advise it as to the attitude which should be adopted by the British representative at the proposed international conference on load line. After a thorough investigation by this committee of all casualties over a period of years it reported that in no case could any of these losses be attributed to insufficient freeboard. Our own subcommittees following the same procedure in the case of coasting vessels have investigated all casualties for a period of years and also report that no case has been found where the loss can be assigned to lack of freeboard.

Investigations are now being made to determine what effect the so-called "Blue Book" load line would have on typical coasting vessels. As far as the work has proceeded the indications are that few, if any, of these coasting steamers would be adversely affected and that the majority would perhaps be allowed a greater draught than they are now operated at, on account of their high side and the limited depth of water in some of the harbors. Further investigation, however, may be necessary to determine if there are exceptions to this preliminary conclusion. It is believed that when the coastwise owners become better acquainted with actual facts and conditions relating to the questions as affecting their vessels, all opposition will be withdrawn.

The advantages of a uniform load line for ocean-going cargo steamers is better understood and appreciated by owners and operators. Practically all maritime countries except the United States have either adopted the British Board of Trade Freeboard

Tables or tables in close agreement and given them the force of law. This, in a measure, establishes an international load line between those nations and facilitates the interchange of courtesies by one country recognizing assignments made by another which often safeguards a vessel against detention when loading in a foreign port. While the United States has not adopted the British Board of Trade Freeboard Tables, it may be said that the United States Shipping Board, Emergency Fleet Corporation, has done so inasmuch as it requires an assignment according to these tables, on all of its veessls. No time should be lost in giving this the force of law pending the next international conference on load lines.

PROF. HERBERT C. SADLER, *Member of Council:*—As a member of the present load-line commission, perhaps it is not proper for me to anticipate the report of that commission. One or two points have been raised in the paper which I think could properly be discussed in this meeting.

Amongst the essential considerations determining freeboard are the conditions of service. As we all know, the conditions of service of ocean-going vessels and coasting vessels, and vessels on the Great Lakes, particularly, are entirely different, and the standard of strength that is sufficient for ocean-going work will, of course, be far in excess of that necessary for some of these other types of vessels. That question is being investigated very fully just now for the Great Lakes and coasting vessels.

That brings up the subject of strength of ships generally. You may recall that a British report is mentioned in the paper which recommends a certain standard of strength. Personally, I have always felt that is a mistake. We have certain recognized registry societies today, and their standards of strength, I think, could well be taken as those necessary for freeboard. It rather adds another annoyance to the designers of ships to introduce a factor of strength in connection with freeboard. You can see what would happen. You might work out an I/Y factor for the midship section suitable for the freeboard and then submit that section to the registry society, and the first thing you know it would come back covered with red ink. Probably a number of the details of that section you had designed might not be accepted by the registry society, and yet it would be perfectly satisfactory from the freeboard standpoint.

I think it was a mistake in the British Committee to lay down any standard in that way. The simpler method would have been to accept the standard registry societies' rules, because, after all, there is not a very great difference in this respect between all the principal well-known societies today. Somebody might suggest that in the future there would be nothing to prevent a new registry society coming out with less stringent requirements. I think that can well be safeguarded if the rules call for a standard equal to that of existing societies at the present day.

It is rather unfortunate that there should be the feeling amongst certain shipowners and shipbuilders that the establishment of a load line is going to work a hardship upon them. As a matter of fact, if you carry the thing out to its logical conclusion, certainly judging from past history, the result is just the opposite. There may be a few people who are overloading ships today, but I think there are very few who might suffer, but in the long run the shipowning business is going to gain by proper regulation. The question of insurance is surely going to be affected as the probability of loss is cut down by reasonable legislation.

THE CHAIRMAN:—We would like to have some further comments on this paper. If there is no further discussion, we will ask the author to reply to any comments that were made.

MR. ARNOTT:—I wish to thank the members who have taken part in the discussion. Mr. Rigg has made out an excellent case for the establishment of a legal load line. Although at the present time this question of compulsory freeboard legislation is exercising the minds of some of our shipowners, I do not know of any naval architect or shipbuilder in this country who is not in favor of a definite standard of freeboard being laid down for merchant vessels. As Mr. Rigg pointed out, it has become customary for new vessels to be paid for at so much per ton deadweight and, if this basis of payment is to continue, statutory freeboard regulations administered by an impartial authority are necessary if only to place designers, builders and owners on a fair commercial basis.

While it is true, as Mr. Rigg pointed out, that any well-designed structure will stand a reasonable overload without danger of actual collapse, continued and repeated overloading is dangerous especially for a ship structure where the overloading may coincide with abnormal weather conditions. Two tankers which had been operating successfully for years recently foundered, the structure failing to meet that particular combination of circumstances and conditions existing at the time of the casulties as to weather, nature, extent and distribution of the cargo loads.

Mr. Rigg very properly emphasized the importance of draught as a basic factor in the design of a ship's structure, and I think he will agree that in the construction rules of the American Bureau of Shipping the load draught is not obscured but has its proper place as a factor to be taken account of in the determination of scantlings.

With reference to the question of a light load line raised by Mr. Saunders, there is no doubt that adequate ballasting arrangements are advisable under modern trading conditions where vessels are unfortunately compelled to make long voyages in light condition, and a stipulated maximum freeboard would certainly tend to reduce the amounts of the bills for damage repairs to the flat of bottom forward. In this connection, it may be of interest to mention that a bill has been recently introduced by the Labor Party of the British Parliament which contains the provision that every vessel now marked with the statutory deep load line must in addition be marked with a light load line which must be kept submerged when at sea.

I agree with Mr. Frear that owners of coastwise steamers have nothing to fear from a compulsory load-line bill. As a matter of fact, the majority of modern coastwise freighters would under the existing freeboard regulations be allowed to load deeper than their present service draughts. While the older coasting vessels with their light hurricane decks and huge cargo doors are not exactly ideal from the point of view of structural strength and can only be considered suitable for coasting service, some of the more modern coasting vessels are quite up to the standard of strength required by the proposed rules of the British 1915 Load Line Committee for unlimited service in association with their present operating draughts.

Professor Sadler in his remarks questioned the advisability of laying down an international standard of strength for freeboard, and this particular feature of the 1915 Load Line Committee's report will probably come under the fire of criticism at any international conference on load lines for the reasons mentioned by Professor Sadler. The freeboard of special types such as Great Lakes vessels was without the scope of this paper,

but it is obvious that an essential consideration to be taken into account in determining the freeboards of these special types is that of the conditions of service which naturally have their effect on strength standards. For instance, service on the Great Lakes is essentially a summer service, where the weather and wave conditions encountered are not to be compared to what an ocean-going vessel may expect to experience in winter in the North Atlantic. Considerations such as the provision of adequate protection of openings are of importance even for lake vessels, as will be admitted by those of us who have observed some of these lakers in which the only obstacle to a heavy sea coming aboard and filling up the after end of the vessel is a wood deck house of light construction.

THE CHAIRMAN:—We all appreciate the amount of work put on this paper, and the thanks of the Society will be extended to Mr. Arnott.

The next business is Paper No. 9, entitled "Recent Advance in Oil Burning," by Mr. Ernest H. Peabody, Member.

MR. PEABODY:—I shall not attempt to read this paper, but merely point out some of the salient points; in fact, the most interesting part of the paper is in the tables, which consist of figures and make a very dry sort of reading.

I take the position that the competition which our Navy Department is meeting is a great asset to us, not only along military lines but also on lines which have proved to be of great advantage in commercial life. We have seen it in many instances, and I think the recent advance in oil burning is another case in which the experimentation of the Navy is pointing the way for the future.

I am greatly indebted to the Navy Department for the privilege of including the data in this paper, and especially appreciate it on account of the fact that some of it is so recent that the official reports have not yet been made. Thanks to the cooperation of Mr. Kain, I was able to delay the manuscript until I got some figures that were made only in the middle of September, which is somewhat after the date, I believe, that the Committee on Papers likes to have manuscripts.

I have included some figures on the number of United States oil-burning vessels which I received through the kindness of the American Bureau of Shipping, and it is very interesting to note that about 56 per cent of the ocean-going self-propelled American fleet is using oil fuel.

Since the paper was written, I have seen some figures in Lloyd's Report, showing that in the world's tonnage the advance in the percentage of vessels using oil was from 10.5 to 16 and a fraction this year, so that increase in the oil-burning installations has been very rapid.

The work of the Navy Department in oil burning has recently been towards larger units, and not only oil burners but in boilers; so looking at the future, in the light of experience in the past, I believe that indicates the pathway we are taking now in regard to oil burners.

It was only a few years ago that 500 pounds of oil per burner per hour were considered a very creditable maximum. However, tests recently made show that more than a ton of oil has been burned per hour per individual burner with excellent efficiency.

Now, in connection with large units and the use of oil fuel for high capacity, I think nothing is more illuminating than the tests made about a year and a half ago at the fuel oil testing plant at the Philadelphia Navy Yard on a White-Forster boiler composed of one-inch tubes, using oil burners of the mechanical atomizing type. I have referred

in the paper to the high capacity run, which I think stands as the world record for high boiler capacity, and high burner capacity with good economy. The oil burned per square foot of heating surface was 1.5 pounds. The combustion was excellent, as shown by the fact that more than 76 per cent efficiency was obtained, while the output of the boiler, figured in pounds of water evaporated per square foot of heating surface from and at 212 degrees, was nearly 23 pounds.

As many of you know, the ordinary land horse-power value calls for an evaporation of 3.5, and that shows what can be done with a large unit with mechanical atomizers, using oil. In that test there were eleven burners each of which burned more than 1,000 pounds of oil per hour, but last summer the Babcock & Wilcox Company brought out a larger type of the same design of burner in which they were able to obtain a burner capacity of 1,500 pounds of oil per burner per hour, with six of the burners in operation in one boiler.

The next step in development was indicated in a test of the burner which I will describe later, a wide-range mechanical atomizer that was installed under the Normand boiler at the Fuel Oil Testing Plant. In this test more than 1,800 pounds was burned per burner per hour with good efficiency. This performance was followed in September by the tests I referred to with a larger size of the same type of Babcock & Wilcox burner. More than a ton of oil per hour per burner was burned, the maximum figure being at the rate of 2,287 pounds for half an hour's run.

While the evaporation figures show a little less efficiency for these large units at the same capacity than a larger number of smaller units, it must be realized that the advance is recent, and all these tests are really in an experimental stage. We have seen repeatedly, and we shall see again, a small number of large units beat out the larger number of small units.

Plate 63 shows a chart where the highest curve was obtained with eleven burners and the second from the bottom with sixteen units of smaller size. In the test of the Normand boiler, three burners were used against about eleven in the previous test. So that while I say that the tests shown here do not actually give the same efficiency as a larger number of small units, the day will come when the efficiency will be maintained with the smaller number of very large units. I feel that that is the trend of development.

As an example of what the large unit means, note that the battle cruisers, which are now being laid down to develop 180,000 horsepower, were only a few months ago to have fifteen burners per boiler. On the basis of the 1,800-pound test, it would only require seven burners per boiler, and when it is considered how this simplifies the piping and saves the necessity for the care of all of these small units, one will appreciate what this increase in the capacity of the burner means when dealing with large powers. While fifteen burners were assigned to the boiler, I think it is questionable whether they could have been put on the front. I doubt if there would be room enough to do that, but there is ample room for the seven burners.

I have referred, at the end of the paper, to a question which has always interested me, and in which I regret to say there has been no advancement, and that is the measurement of viscosity.

As you all know, there are a half dozen viscometers and scales, and I have recently seen a reference to a new complication, namely, "Engler seconds." I am fortunate in being able to present data showing some relation between the Saybolt seconds and the Engler degrees, which is probably accurate enough for practical purposes.

In closing I want to describe the Fisher burner which the company I am now with is experimenting on in connection with the Navy Department. Commander Fisher, now fleet engineer of the Atlantic fleet, is the inventor of this principle. During the war, while Commander Fisher was at the Bureau of Engineering in charge of the submarines, he became very much interested in the internal combustion engine. Being familiar with the mechanical atomizer as used in the Navy, he conceived the idea of spraying the oil into the internal-combustion engine by means of this mechanical atomizer, but when he tried it he could not make it work, and the reason is quite apparent.

As you alternately close and open the orifice, you stop the flow of oil to the burner and so lose your atomizing effect of the oil. I am assuming that you are familiar with the principle on which the mechanical atomizer works. There is, as you know, the air and steam atomizer which uses an atomizing medium for spraying the oil. In the mechanical atomizer the oil is delivered through small tangential channels to the central chamber in the burner, where it receives a rapid whirling motion and issues from the orifice in the form of spray, induced by centrifugal force.

In all the mechanical burners which have been used heretofore, all the oil, which enters the central chamber through these small tangential slots, finds an exit through the orifice of the burner, and in changing the capacity of the burner the only means available is to vary the pressure under which the oil is delivered to the burner. For instance, a burner designed for a capacity of 800 pounds per hour, avoirdupois, at 200-pound pressure can be operated also at about 50-pound pressure, in which case it will deliver about 400 pounds per hour; but you cannot reduce the capacity materially below 400 pounds because through friction and other causes you then begin to lose your rotary motion and spraying action.

In connection with this experiment in the internal-combustion engine, Commander Fisher conceived the idea of passing some of the oil instead of making it all go through the orifice. Thus, if he carried it back through the by-pass, he could close the orifice and still keep the oil moving in the central chamber. I would like to ask you to look at Plate 68.

This shows the form of the Fisher burner, which has been used in the Navy. I am indebted to the Navy Department for permission to publish this data. The oil enters through the annular space, between an inner and outer pipe. It goes through openings in the copper disc, passes along the longitudinal slots in the plug shown to the left, and enters the central chamber through the tangential channels shown in the end view of the plug, and there the oil receives the whirling motion. If the outlet leading from the spindle shown in the center is closed, the burner works in the same way as the ordinary mechanical atomizer. All the oil then finds exit through the orifice.

If the spindle is drawn back, some of the oil can be passed around the spindle into the annular space, between that and the pipe next to it, so that by regulating the pressure of the oil to the outlet, or in this case by varying the position of the spindle, the amount of oil by-passed can be varied. The effect of this is, as applied to the internal-combustion engine, that he can stop the flow through the orifice entirely and then open the valve and stop again and again open, and during all these processes, still keep up the atomizing action. The effect of this on a continuous flow burner under a boiler is that you can go much below the former limit of 50 per cent of the maximum owing to the fact that you are continually keeping up the maximum whirling effect in the central chamber. The rotary effect is the same, because the maximum amount of oil is entering

the chamber, and you can take away through the by-pass a large proportion; our experiments show from about 80 to 90 per cent.

For a fluctuation load we believe that this burner has a very useful field, and we will be able to cover wide ranges with a few large unit burners. I may add that the work referred to here in the Navy Department seems to me to be the only marked advance in oil burning. We have seen some very large vessels fitted up for oil, but they are following along the precedent of some years back, and there is still a large field in which good economy with higher capacity may be obtained with oil.

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RECENT ADVANCE IN OIL BURNING.

By Ernest H. Peabody, Esq., Member.

[Read at the twenty-eighth general meeting of the Society of Naval Architects and Marine Engineers, held in New York, November 11 and 12, 1920.]

In the few years which have passed since a paper on the subject of oil fuel was presented before this Society, the great World War has been fought and won by the powers opposed to barbarism—won so recently, in fact, that the question "who will win the peace" is still debatable.

During this war the use of oil for the generation of steam has assumed an unprecedented importance, and I do not think it wide of the mark to say that the next great war will be won by the nation most plentifully supplied with oil fuel; at least it cannot be won by any nation whose supply of oil will not successfully meet a stupendous demand. It must be hoped that our own government will immediately and adequately recognize this condition.

But it is not with the supply of oil that I intend to deal, except to say that I am reliably informed by those in a position to know that the American driller, the American tanker and the American dollar will assure this country of the oil it needs, if they are properly backed by the American people.

In speaking of the next war I may be charged with being a reactionary. I hopefully share the view that some day international warfare may be controlled to a degree proportionate to the success with which private warfare has already been reduced in civilized nations, but the whole scheme of nature shows that to survive one must be prepared to fight, and preparedness to-day largely signifies the ability and readiness to organize and use natural forces to the limit of skill and ingenuity.

Regret it as we may, the competitive necessity of warfare inspires more rapid scientific progress than times of peace. It is not strange, therefore, that we may look to the military arm of our government for the first signs of advance in the use of natural forces. In time of war they face the keenest competition of other nations, and in time of peace this competition is in no way lessened even if it be potential in character. Nowhere is this leadership better illustrated than in the now universal use of the mechanical atomizer on oil-burning ships. First exploited in vessels of war, it has found a fertile field in the merchant marine.

The United States Navy is again pointing the way to future development in oil burning. In cooperation with leading mercantile interests who are manufacturing equipment for the Navy, they are experimenting not only in the use of heavy viscous oil but are demanding larger units and greater individual capacity both in boilers and in oil burners. I predict that this increase in size and rate of forcing will mark the trend of future development in the merchant marine. To-day there has been no marked advance along these lines in the merchant service, not-

withstanding the fact that the number of oil-burning ships has been enormously increased and our imagination has been stirred by the fitting up for oil fuel of some of the largest transatlantic liners.

I am indebted to the American Bureau of Shipping for permission to use the following estimate of the number of oil-burning vessels as compared to those burning coal. Under date of September 15 they state:—

"1. For the coal and oil-burning data of vessels now operating, the following has been compiled from various sources, none of which gives tonnage figures. It is to be assumed, however, that authentic tables, being used in comparison with those that are incomplete, will give a good knowledge of the information desired.

"2. United States Shipping Board vessels, which make up the bulk of American vessels, are daily changed from coal burners to oil.

"3. On August 1, 1920, there were documented in the United States the following number of steel and wooden vessels, of 500 gross tons and up: 2,622 vessels of 10,535,922 gross tons. Of these the U. S. S. B. owned 1,662 vessels of 7,074,971 gross tons, thus leaving to private owners 960 vessels of 3,460,951 gross tons.

"4. No comprehensive data being available as to the fuel of these ships we will make a comparison with those facts available, as of January 1, 1920, pertinent to U. S. S. B. vessels only, *i. e.*:—

	Vessels.	*Per ce t.*
Coal burners	841	56.75
Oil burners	635	42.5
Non-propelled	7	.75
	1,483	

"Applying this ratio to the present Shipping Board holdings of 1,662 vessels, we find:—

	Vessels.
Coal burners, approximately	945
Oil burners, approximately	705
Non-propelled, approximately	12
	1,662

"Of the 960 privately owned ships, the same method must be applied. The American Bureau of Shipping report of September 1, 1920, on steel vessels building in the United States to private account, having authentic figures relative to the fuel of these vessels, will be of value. It follows:—

	Vessels.	*Gross tons.*
Cargo, coal	20	38,171
Cargo, oil	101	400,455
Cargo, coal and oil	13	74,567
Cargo, non-propelled	109	82,350
	243	595,543

	Vessels.	*Gross tons.*
Tanker, coal	...	.....
Tanker, oil	121	830,964
Tanker, coal and oil	...	.....
Tanker, non-propelled	19	11,991
	140	842,955

Grand total, 383 vessels; 1,438,498 gross tons.

"6. Summarizing, we find:—

	Per cent.
Coal burners	5.2
Oil burners	58
Coal and oil burners	3.4
Non-propelled	33.4

"7. Applying these percentages to the 960 ships in question, we find:—Coal burners, approximately, 48 vessels; oil burners, approximately, 555 vessels; the balance being mainly non-propelled.

"8. It is interesting to note at this point that of the tonnage (steel) now building to private account, more than 58 per cent is made up of tankers.

"9. Recapitulation of the above data brings out the following:—

Operating:

	Vessels.	*Per cent.*
Coal burners, approximately	993	37.8
Oil burners, approximately	1,260	48
Balance (mainly non-propelled)	369	14.2
American vessels	2,622	

"(Signed) CROWLEY,
"*Statistician, ABS.*"

Thus, while very large vessels are using oil fuel successfully and from the estimate of the American Bureau of Shipping it is evident that nearly 56 per cent of the present American self-propelled merchant fleet consists of oil-burning vessels, it is probable that in not one of these ships is there a boiler containing more than 4,500 square feet of heating surface in the individual unit and no single oil burner capable of atomizing over 600 pounds of oil per hour, while the oil burned per hour per square foot of heating surface under forced-draft conditions will not equal one-half pound.

In other words, the present boiler and oil-burner practice in the merchant marine is following closely along the very conservative lines of previous custom, and the full value of the use of oil fuel has not yet been approached. Owners and designers have been content with the many great savings and advantages of oil fuel without perhaps realizing all its possibilities in the line of increased capacity without serious loss in economy.

There are many reasons for this. In the first place a large number of vessels now burning oil were originally designed for coal, and in others, which were designed to burn oil, the old proportions applying to the coal-burning ships have been adhered to, in many instances probably because it is feared that a return to coal fuel may some day become necessary. Furthermore, in the large majority of instances the vessels are fitted with Scotch boilers, and the above figures mark practically the limit to which such boilers may be forced with safety. The same statement will apply also to water-tube boilers until such time as more consideration is given to the type of condensers and the size and design of evaporators.

In marked contrast to the merchant marine, the Navy Department is making rapid advance of a most important character. It is now using water-tube boilers entirely in fighting ships, with improved condensers and evaporators. Boilers are being constructed containing more than 11,000 square feet of heating surface, while in the matter of size and capacity of oil burners, within the last few weeks successful tests have been run at the Fuel Oil Testing Plant at the Philadelphia Navy Yard in which more than one ton of oil was atomized per hour in the individual burners.

If then, as I believe, the work of the Navy is a forecast of what will eventually be the trend of development in the merchant marine, not only will considerable improvement be made in oil-burning equipment, but the Scotch boiler, with its enormous weight, poor circulation and unsuitability to high pressure, will become a thing of the past. This type of boiler has long been tottering on its last support of prejudice and the reactionary belief that it is impossible to keep salt water out of the feed system. We shall have, in its place, large, safe, light, economical water-tube boilers, installed in closed fire-rooms, burning oil and operating under forced draft with a few large-unit mechanical atomizers.

The remarkable economy of the properly designed small tube express-type water-tube boiler at high rates of forcing with oil fires was demonstrated not very long ago in some notable tests at the Fuel Oil Testing Plant. The tests were conducted under the direction of Lieut. Commanders A. M. Penn and W. R. Purnell, U. S. Navy, and their staff of trained assistants. The boiler was of the White-Forster design, built by The Babcock and Wilcox Company, and contained 7,565 square feet of heating surface and 753 square feet of superheating surface, the furnace volume being 751 cubic feet. Three different types of air registers were used:—the Bureau of Engineering forced-draft design (of which there were sixteen units installed), the Bureau of Engineering natural-forced-draft design (eleven units), and what was then known as the Peabody register manufactured by The Babcock and Wilcox Company, designed for either forced or natural draught and of which there were also eleven units. All three registers were tested successively with bureau burners. Final tests were then made with Peabody burners with Peabody air registers only.*

*This equipment is now designated as The Babcock and Wilcox Air Register and The Babcock and Wilcox Mechanical Atomizer.

Through the courtesy of the Navy Department I am able to include in this paper the official results of the last two series of these tests in the tables, Plates 64 and 65, and I have taken the liberty of giving the plotted efficiency results of all four series in Plate 63.

It will be noted that in the highest capacity test of June 10, 1919, there were eleven burners in operation, each atomizing 1,032 pounds of Navy Standard Oil (25.5 Baumé) per hour, giving a consumption of 1.5 pounds of oil per square foot of heating surface per hour. The evaporation of water per pound of oil from and at 212° F. was 15.14 pounds, giving an evaporation of water per square foot of heating surface per hour of 22.73 pounds from and at 212° with an efficiency of 76.15 per cent. The air pressure in the closed fire-room was 9.5 inches and the rate of combustion per cubic foot of furnace volume reached the very high figure of 15.12 pounds of oil per hour. It is believed that this test stands as a world's record for efficiency at high boiler and furnace capacity.

In the burning of oil there are many things more important than the mere rate of combustion as measured in pounds of oil per square foot of heating surface. The latter applies rather more to the boiler, and the ultimate capacity is a function of efficiency in heat absorption as well as the amount of oil burned. But it is interesting to note that tests have been made in England by Babcock & Wilcox, Ltd., in which the rate of combustion practically equalled 1¾ pounds of oil per square foot of heating surface per hour. I am indebted to The Babcock and Wilcox Company of New York for the following memorandum concerning these tests. Under date of September 9, 1920, my friend and former colleague, Capt. Walter M. McFarland, writes:

"The test of a Babcock and Wilcox boiler was made at Renfrew in the early part of 1917. This boiler was composed almost entirely of small tubes and was entirely experimental, as a first step in a series of studies.

"Six tests were made, in which the oil was burned at a rate of 1.2 pounds per square foot of heating surface per hour, or higher; one test was at the rate of 1.53 pounds with an air pressure of 2 inches, and the last was at the rate of 1.74 pounds with an air pressure of 2.08 inches. In the former case the baffling was of checkerwork on the top of the tubes, while in the latter there was no baffling at all." The oil burned per cubic foot of furnace volume at the highest rate was 11.64 pounds per hour.

In a paper before the Institution of Naval Architects presented by Mr. Harold Yarrow some years ago, reprinted in the Journal of the American Society of Naval Engineers for May, 1912, a test on a Yarrow boiler was reported in which the figures 1.94 were given as the rate of combustion in pounds of oil per square foot of heating surface per hour. It is to be noted, however, that in this test a damper in the uptake over one bank of tubes was closed and that the above high rate of combustion was reported on the assumption that only half of the heating surface was making steam. This assumption is, of course, unjustified, and it appears that the highest rate obtained in the Yarrow test was 1.24 per square foot of heating surface.

Impressed as I am with the importance of naval experiments, I have made an effort to secure some information as to what was being done in the development of oil burning by the British Admiralty. In this I have not been successful, but I am informed that a paper dealing with this subject is to be presented by Sir George G. Goodwin, Engineer in Chief of the British Admiralty, before the Institute of Naval Architects next February.

Returning to the work of the United States Fuel Oil Testing Plant now under the command of Lieut. Commander Purnell, it seemed at first sight that 1,032 pounds of oil per burner per hour was rather a notable achievement. It is interesting to state, therefore, that this record has already been materially outstripped. Early in the past summer The Babcock & Wilcox Company offered six air registers of their standard design, but of an enlarged type, with mechanical atomizers capable of atomizing 1,500 pounds of oil per unit per hour. These were tested by Commander Purnell under the same White-Forster boiler already described. The six burners successfully sprayed over 9,000 pounds of oil per hour under conditions similar to those pertaining to the U. S. scout cruisers now under construction. The oil per square foot of heating surface was approximately 1.2 pounds, and the oil per cubic foot of furnace volume per hour about 12 pounds.

This record was exceeded in August in a test of the Normand boiler at the Fuel Oil Testing Plant. This boiler contained 4,500 square feet of heating surface and 487 cubic feet furnace volume and was fitted with three of the recently developed Peabody-Fisher wide-range mechanical burners and new Peabody air registers. Crude Mexican oil of 13.3 Baumé was used and the three burners each atomized over 1,800 pounds of oil per burner per hour, without smoke, carbon or other objectionable conditions. The oil burned per square foot of heating surface was 1.2 pounds per hour and per cubic foot of furnace volume 11.15 pounds per hour.

The following letter from Commander Purnell enclosed the results of the tests as given in Table II, Plate 66:—

"The Peabody Engineering Corporation,
331 Madison Avenue,
New York, N. Y.

Attention of Mr. E. H. Peabody, President.

"Gentlemen:—

"I am enclosing summary of results of tests conducted on the Normand boiler, which was equipped with three Peabody air registers and three Peabody-Fisher wide-range fuel-oil atomizers.

"The Normand boiler has not been used for testing purposes for approximately five years and is not equipped with the latest appliances and instruments for securing test data. However, I believe the final calculations can be taken as representative of what can be expected from the boiler.

"While the tests were not official in so far as they were not run to meet specific

requirements, the same procedure and methods of testing were followed as are customary on an official series.

"Yours very truly,

"(Signed) W. R. PURNELL,

"*Lieutenant Commander, U. S. N.*"

It is but fair to say that this rather remarkable capacity test was in a measure unexpected, as the original intention was to burn 1,500 pounds of oil per hour per burner. Also, while the boiler and furnace efficiency is low compared to results in the White-Forster tests, the results are better than the previous record on the Normand boiler as reported by Lieut. Commander (now Captain) John J. Hyland, formerly in command of the Testing Plant. Thus at the rate of .455 pound of oil per square foot of heating surface per hour the previous efficiency result was 74.5 per cent, while the present record is 80.67 per cent. No test was run previously at a rate higher than 1.05 pounds of oil per square foot of heating surface per hour, but the efficiency curve extended to the 1.2 pounds mark shows 64.5 per cent, while the present record is 68.84 per cent.

A new record in oil per burner per hour was established on September 17 by Commander Purnell when three of the six Babcock and Wilcox units were tested, the burners being supplied with larger tips. The oil used was 19.9 Baumé. The test was of one hour duration but followed several hours' steaming during which the individual burner capacity was pushed up to 2,000 pounds per hour. The record for the hour shows 2,238 pounds, while in the last half-hour the rate of combustion per burner reached 2,287, a safe margin over one ton of oil per burner per hour.

Owing to the three burners only being in operation in the large furnace it is to be noted that the oil burned per cubic foot of furnace volume was reduced to 8.9 pounds per hour.

I am again indebted to the Navy Department for permission to publish these remarkable tests (see Table III, Plate 67). The privilege is especially appreciated for the reason that the official report has not yet been completed.

These tests are so very recent that it is not yet possible to foretell their precise significance, but it is worthy of note that six months ago fifteen burners were being considered for each of the sixteen boilers of the battle cruisers now being laid down. On the basis of this last performance only five would be required, and seven would be a safe estimate. To develop 180,000 horse-power with 80 or even 112 oil burners is a very notable performance, but there can be no doubt about its being entirely possible with the equipment which has been perfected in this country. We shall no doubt hear the time-honored warning against putting too many eggs in one basket, but nevertheless I am convinced that the large unit, both in boilers and oil burners, has come to stay, particularly if it carries with it efficiency, accessibility, simplicity, safety and flexibility, which it most assuredly does.

I do not wish to be understood as saying that I think that oil burners atomizing a ton of oil per burner per hour will be immediately adopted for naval service, far less for the merchant marine, but the trend of modern development is certainly

in the direction of larger units and the higher capacities made possible by the use of oil fuel.

Referring to the matter of flexibility, I think I am justified in discussing another recent development which I believe marks a distinct advance in the oil-burning art. It is well known that the mechanical atomizer of the usual design has a decided limitation in range. Unlike the steam atomizer, it cannot be operated at low capacity if it is designed for a high one. The lower limit with the oil pressures ordinarily used is approximately one-half the upper. The reason for this is very simple. The burner depends for its atomizing effect on centrifugal force induced by giving the oil a rapid whirling motion in the central chamber of the tip. This whirling motion, in turn, is produced by delivering the oil to the central chamber through eccentric passages, or small channels substantially tangential to the walls of the chamber. All the oil which enters the chamber through these channels finds an exit through an orifice concentric with the axis of rotation. Thus as the capacity of the burner is reduced by the obvious method of reducing the oil pressure, there is a lower velocity through the channels and friction losses become more apparent, until finally the spraying effect is reduced till the oil issues from the tip in a solid stream or in a spray too coarse and uncertain to give the desired results. In burners of the usual type this point, as stated, is reached at about one-half the maximum capacity of the burner.

In atomizers of the Thornycroft type, in which the size of the tangential channels may be reduced by an adjustment in the burner, the velocity through the channels is kept up to a large extent at low powers, and the range of these burners is much greater than in those of the usual type. But it has been found difficult in practice to make individual burner adjustment under service conditions, and this method has never become popular, even if it be assumed that a sufficiently wide range in capacity is secured thereby, which is questionable. The only alternative, therefore, with burners of the ordinary design is to shut off some of the burners at low powers, or to change the size of tips, or to do both, which is not unusual. Both these methods are open to objection.

Another and radically new plan, however, has been devised which promises very satisfactory results. This consists of maintaining the whirling motion in the central chamber of the tip undiminished whatever may be the capacity desired of the burner; in other words, instead of delivering through the orifice all the oil which enters the burner chamber, as is invariably done in all other mechanical atomizers, a part of the oil supply is diverted or by-passed from the central chamber and returned to the pump suction, and the actual effective capacity or amount of oil which is sprayed into the furnace depends merely on the proportion of the oil that is by-passed to the total amount entering the burner. Thus, as the amount of oil entering the central chamber through the tangential slots remains at all times at the maximum, the whirling of the oil and the atomizing effect remains also at a maximum, so that a perfect spray is secured whatever the capacity, at low powers as well as at the highest rate. By this method the range of the atomizer is very greatly increased, and, instead of being limited at the low powers to 50 per cent

of the maximum, a good spray can be secured all the way down to about 10 per cent of the total.

This method of spraying is the invention of Commander J. O. Fisher, U. S. Navy, now fleet engineer of the U. S. Atlantic fleet. He first applied it in an attempt to spray oil into the cylinders of an internal-combustion engine, where, owing to the intermittent action required, the ordinary mechanical atomizer was an entire failure. By introducing the by-pass a continuous flow was secured, even when the regular outlet orifice was closed. The application of the principle to atomizers for boiler work followed, with the result that a single burner without change of tip could be used over a greatly extended capacity range. The Navy Department has applied the Fisher burner with great success to small boilers of the launch type, where its wide range enables them to operate the boiler with only one burner. I have shown the navy design of this burner in Plate 68. The company with which I am now connected has developed a modified form of the Fisher burner with which some interesting experiments have been carried out with the cooperation of the Navy Department. It is believed that the wide range in capacity will make the burner especially useful for fluctuating loads or in any installation where it is now necessary to change the tips or close off a portion of the burners.

It may be of interest to state that the burners having a maximum hourly output of 1,820 pounds which were tested at the Fuel Oil Testing Plant were successfully operated at 250 pounds, while smaller tips designed for a maximum of 800 pounds were operated at a minimum of 80 pounds per hour. In our design we have dispensed with the spindle shown in Plate 68, and we control the capacity entirely by an external valve in the by-pass return line, so that by the manipulation of this valve any number of burners can be simultaneously increased or decreased in capacity over the entire range. Also, the tangential channels and the outlet orifice are in a single piece.

A matter closely allied to the use of oil as fuel, in which there has been no advance and, if anything, a retrograde movement, is the measurement of the viscosity of oil. Unless I am grossly uninformed, we are no nearer a universal standard method of designating viscosity than ever, while, on the contrary, not only are new viscometers and new scales being devised, but there is a tendency in certain quarters to use Engler "seconds" in the same way that Saybolt seconds are used, instead of dividing the time of outflow of the oil in the Engler viscometer by the time of outflow of an equal amount of water. The latter gives the true Engler scale, as we have been taught to regard it, and the use of Engler seconds only adds to the confusion.

It is probable that if the Saybolt viscometer had been obtainable by anybody outside the large oil companies ten or twelve years ago when the mechanical atomizer began to be used extensively, we should not now be using the Engler instrument at all in this country, notwithstanding its extensive use in Europe. But some method of determining viscosity became necessary and, as usual, the Germans were on the job and supplied us with Engler instruments.

Various methods of interconversion of the two scales have been suggested, but the problem is not as easy as it looks, as the conversion factor is not constant throughout the viscosity range and it also varies with the character of oil. These differences are perhaps more academic than of real importance, as approximation to the viscosity results will usually serve.

Roughly speaking, the Engler degrees are 1/36 of the Saybolt seconds, *i. e.*, $S = 36E$, or $E = \frac{S}{36}$ where S = Saybolt seconds and E = Engler degrees (not seconds).

I am fortunate in being able to present a chart (Plates 69 and 70) prepared by Mr. R. C. Brierly, chemist at the Fuel Oil Testing Plant, showing the relation of the two scales by actual test of samples of the same oil at the same temperature in the two types of viscometer. The solid line in the chart represents the ratio $E = \frac{S}{37{\cdot}5}$, which is sometimes used for approximate determinations. The dotted line connecting the various points determined by test favors the ratio of 36 to 1.

Mr. Brierly states:—

"I am sending you enclosed the graphical representation of the theoretical Engler-Saybolt ratio and points plotted from our determinations. It is unfortunate that these ratio tests were not run at one or two certain temperatures, as temperature variation is undoubtedly a most important factor. However, I have noted oil temperature beside each plotted point.

"As a matter of interest I ran a sample of 'Binderine'—paper pulp refuse—on both machines for comparison. I have shown it on the curve. Two of the most important references for the 1:37½ ratio are the U. S. Navy Oil Fuel Specifications of 1917 and H. G. Nevitt in *Chemical and Metal Engineering* for June 23, 1920, page 1171."

TABULATION OF OILS USED IN SAYBOLT-ENGLER COMPARISONS.

Number.	Name of oil.	Sp. G. at 60° F.	Flash point °F.	B. T. U. lb. actual.	Remarks.
S–144	From U. S. S. Sub. K–1.....	.8581	185	*19588	Diesel oil.
S–44	J. H. Ref. Co., 3/14/18.....	.8940	185	*19322	
S–41	U. S. S. Mayrant.....	.8922	160	*19334	Used by Captain Bixby in trip to Norfolk.
S–126	U. S. S. S–2.....	.8930	225	19226	
S–45	U. T. L. X. 11555.....	.8892	180	*19355	
S–47	W. F. Boiler test, 4/2/18.....	.8917	200	19558	Mixed tanks.
S–122	Lub. oil G. Sub. U–164.....	.9301	350	.....	Grayish black loaded with animal oil.
S–124	Lub. oil G. Sub. U–164.....	.9294	360	.....	Muddy color smells like whiskey.
S–123	Lub. oil G. Sub. U–164.....	.9313	380	.....	Light oil.
S–139	W. F. test, 4/16/19, Tidewater..	.9137	160	19102	
S–304	Vetol Motor oil.....	.....	405	.....	Light green.
S–205	Bunk "A" of Tex.....	.....	....	.....	
S–203-4	Panuco, 1st car load.....	.9774	†120	18129	
S–13	Texaco Mex. Crude.....	.9760	86	18179	
S–11	Reduced Mex. Fuel.....	.9580	195	19033	
S–104	At. Ref. Bunker C.....	.9635	160	18621	
S–103	At. Ref. Bunker B.....	.9429	160	18994	
S–236	Oil used at Erie, Pa.....	.9610	†185	18564	Jan. 6, 1920.
S–16	"Freeport".....	.9700	142	18216	Mr. Pennycook, Freeport Mex. Fuel Corp., New Orleans, La.
S–278	T. C. X. 4832.....	.9516	†205	*18938	
S–267	Texaco 10.8° Be oil.....	.9945	†185	18682	
S–179	Schutte & Koerting exp. test oil.	.9877	‡	*18722	Sub. by Lieutenant Commander A. M Penn.
S–273	Quimby test oil.....	.9974	180	*18664	Sub. by Wm. E. Quimby, Inc.

Flash point Pensky-Marten closed cup unless otherwise noted.
*Indicates given B. T. U. is calculated instead of actual.
†Indicates Cleveland open cup.
‡Unobtainable on account of expansion.

It is to be hoped that some international standard method of measuring the viscosity of oil will be adopted, reconciling the Saybolt, Engler and Redwood scales and eliminating the periodic appearance of new proposals for determining viscosity. Only then shall we all be able to speak the same language.

DISCUSSION.

THE CHAIRMAN:—Will Rear Admiral Dyson discuss this paper?

REAR ADMIRAL C. W. DYSON, U. S. N., *Member*:—I was very much pleased on learning that Mr. Peabody was to present an article on fuel burning before the Society.

I have been closely associated with Mr. Peabody for many years in the development of pressure atomization for the burning of oil fuel and am thoroughly cognizant with the work which he has done. Mr. Peabody, under the auspices of the Babcock & Wilcox Company, has been one of the few engineers in this country who has devoted himself full-heartedly toward the perfection and the development of the possibilities of burning oil fuel with present pressure atomization. I think it is due to Mr. Peabody that the whirling plate has supplanted the screw thread for obtaining whirling motion of the oil inside of the burners. He is one of the few that encouraged us in fighting to retain this system of atomization in the Navy, while the majority of others were extremely antagonistic to it, and his farsightedness is shown by the results obtained in burning oil with this system today.

He refers in his article to the Peabody-Fisher type of burner. My first experience with this burner was in the case of the boilers developed by the Navy Department for use in the 50-foot steamers. These boilers were required to provide steam for 200 shaft horse-power, and the space in which they could be accommodated was so limited that the length of furnace was only about 4 feet, while the total cubical content of the furnace was about 30 cubic feet. The short length of the furnace necessitated the development of a special burner that would insure oil being consumed within a short distance of the burner tip, for if this were not so, the unconsumed oil at the tip of the flame struck against the end wall of furnace, causing a building up of carbon and a formation of heavy smoke.

After various attempts we tried using the Fisher burner and met with instant success. With this burner in this small furnace we have burned nearly 14 pounds of oil per cubic foot of furnace volume, have evaporated nearly 14 pounds of water per pound of oil, while the smoke formed has been small in amount and very light in color. The burner has the advantage of being able to fix the angle of the flame cone to almost any amount desired up to nearly 180 degrees, and it is this feature which makes it so satisfactory for burning oil in a furnace of short length. As pointed out by Mr. Peabody, it also has another advantage of enormous elasticity in capacity, thus doing away with the necessity of changing burner tips in passing from large capacities to small.

Engineers must soon become educated to the point where they will realize that the successful and economic burning of oil fuel requires quite an education in the art, if it may be so called. The Navy early appreciated this and, with it in mind, developed the Oil Fuel Testing Plant at the Navy Yard, Philadelphia. This plant paid for itself over and over in value received by us from it, and during the war its facilities were utilized to provide what may be called a college course in oil-fuel burning for officers and enlisted men of the Navy in developing crews for the enormous number of destroyers which were being put into commission. In addition to these schools we also had a training school at Fore River and one at Mare Island. As a result the new destroyers, upon first going to sea, always had a nucleus of trained fuel-oil burner men among their crews.

In my opinion Mr. Peabody, in his work, will stand in the first rank of those who in this country have done the most for the successful burning of oil fuel.

MR. W. W. SMITH, *Member:*—Mr. Peabody has given us a very valuable paper on oil burning, especially so as the tests were accurately and thoroughly made.

Flexibility and high capacity are not so essential in merchant vessels, but they are very important in naval service. It is a very great advantage to have flexibility for maneuvering, and especially so for certain services.

The ease of control of this installation is one of its important features, and it may be noted that an entire battery of boilers or burners can be regulated by the operation of one small valve. I was very much impressed when I saw the test of this apparatus and the ease with which the power could be controlled by one man. The flexibility and ease of operation of this apparatus I consider truly remarkable. It has a great range of power, with practically smokeless combustion, which is, of course, an important feature.

I note that these tests were made on a water-tube boiler. I have seen a great many tests on water-tube boilers recently, but practically none on Scotch boilers. It would be valuable to have tests on Scotch boilers also, in view of the fact that a large number of boilers of this type are in use.

Mr. Peabody is to be congratulated on the excellent results and high efficiencies obtained with the types of oil-burning apparatus which he has developed. He has contributed greatly to the science of all oil burning and to the valuable research data in connection therewith. Recirculating the oil from the tip of the burner is, as far as I know, a novel feature in oil burning, and it appears that it should have important advantages where flexibility is an important consideration.

MR. JOHN MARTIN, *Member:*—Mr. Peabody seems to be a pioneer in the marine oil-burning development in this country. His activities seem to have been wholly in this investigation in the naval field, in the operation of naval vessels. The American Bureau of Shipping, however, for the operation of merchant ships has to act as a safety brake, inasmuch as it must prescribe certain regulations for the safe carriage, transmission and burning of oil under boilers, and it has formulated certain requirements for the safe use of oil under boilers. That is a phase of the subject which I do not think the author has covered, and one which I think is of vital importance in the operation of the merchant marine.

MR. HENRY C. E. MEYER, *Member:*—Mr. Peabody is to be congratulated on the able paper he has presented to us. As some of the gentlemen have already remarked, I think Mr. Peabody is about as well qualified to speak about oil-fuel burning as any man I know. I have personally had the pleasure of seeing some of the League Island tests he referred to and was certainly astonished at the remarkable flexibility of the burner he demonstrated there.

While the advantages of great flexibility are of primary importance in naval vessels, I believe they are of sufficient importance to receive consideration in the merchant service.

Many vessels are not operating all the time under full power, and this has particular reference to the case of maneuvering conditions, such as where a ship comes into port or is working in a fog and is slowed down to a considerable extent. In such cases, where you have a large number of burners and have to cut out several of these burners to reduce your power, I think the liability of setting up serious strains in your boilers

is quite obvious, particularly so in the case of Scotch boilers, which, at the present time, represent a majority of the boilers in merchant vessels. If you have to maneuver a ship into port with one burner out of three cut out, you will probably set up serious expansion and contraction strains in the boilers, and any advance made in the ability of the burners to operate efficiently over a wide range is a distinct advantage.

The thanks of the Society are due to Mr. Peabody for the work he has done, and also for the data that he has presented and made available to all of the members.

THE CHAIRMAN:—Is there any further discussion? If not, we will ask Mr. Peabody to close the discussion on the paper?

MR. PEABODY:—I want to thank the gentlemen who have spoken for their very kind comments on the paper. I am sorry that I cannot claim all the pioneering credit in developing the mechanical atomizer that my good friend Admiral Dyson attributes to me, but it is a subject in which I have been much interested, and it has been my good fortune to have done a good deal of work in this field.

Mr. Martin, of the American Bureau of Shipping, raises the question of safety. I did not intend in this paper to cover the whole subject of oil burning, but only to touch on recent advances in the art, and I am sorry if I have given the impression that my work has been entirely in the Navy and that there has been no consideration given to the matter of safety. I have had quite as much experience in the merchant marine as in the Navy, and it happens just now that I am installing some mechanical atomizers in stationary plants on shore.

I do not think that our naval engineers have in any way ignored the principles of safety. We all believe in safety measures—safety first—and welcome any rules or systems which promote that much desired end. But it must be remembered that oil is a safe fuel to start with, and the principles on which safety precautions are based are very well understood today.

So I feel, in giving the palm to the Navy Department for making the greatest recent progress in this line as in many others, we are in no way reducing the factor of safety.

The type of boiler used in naval service is a much less dangerous proposition than the Scotch boiler so frequently met with in the merchant marine, and the large unit is just as safe as the little one. Admiral Dyson suggests that the large unit oil burner in reality adds to the margin of safety by reason of keeping the burner centers farther removed from the boiler tubes, thus avoiding blow-pipe action.

I am very glad that Mr. Martin raised this point as I should not wish to be misunderstood in my views concerning the great importance of safety in oil burning as in any other line of human activity.

I again beg to express my appreciation for the favorable criticism of the paper.

THE CHAIRMAN:—Gentlemen, if we believed some of the pessimists, it will not be long before there is no oil at all, and they will not be pleased with Mr. Peabody's method of burning a great deal more of it, which I have no doubt will be very successful.

The thanks of the meeting will be given to the author for the presentation of his paper, and we will proceed to the next paper, No. 10, entitled, "The Problem of the Hull and Its Screw Propeller," by Rear Admiral C. W. Dyson, Member.

REAR ADMIRAL DYSON:—I think the majority of you gentlemen are thoroughly conversant with the work that I have been doing on the screw propeller for a good many years. I have presented to you in various publications from time to time what I thought were advances; sometimes they were advances and sometimes later on I had to take them back. What I am presenting to you today is my last step, which to my mind is very nearly the final step towards results by which the design of screw propellers can be brought down to a practical designing basis. The future may open up other steps; I do not know. The paper, as I have prepared it, is such that I can only read extracts from parts of it.

Rear Admiral Dyson then abstracted the paper, and during the same said:—

"I want you gentlemen to understand that in most work on screw propellers we find rules for obtaining what they claim will be the successful screw propellers, but we do not find any rules whatever telling us what the successful screw propeller will perform when the ship is heavily loaded, and in conditions of load that become abnormal.

"In my work I was driven into the abnormal conditions, on account of having to handle submarines, towboats, and now and then I would get hold of merchant ships which vary very much from the trial condition down to the seagoing, heavily loaded condition. It is a very easy problem to design a propeller for the submarine when you design for the surface condition only, but when you realize that the submarine has to run submerged at times, you must get a close estimate of the actual revolutions which will occur under this overload condition to prevent overloading your motor. If you get your pitch too high, when you put the full load on the motors, the motors overheat and burn out very shortly, and so I was driven to investigate the unsafe range, while the ordinary investigator in propellers investigates only in the safe range."

In connection with a portion of the report reading, "It has been shown how vessels can be divided into three general types," Admiral Dyson said:—"I know with regular merchant ships which have a high value of K, when they pass over the slip of the first order, and into slips of the second order, they get into cavitation at once. There is heavy increase in K, and also loss due to cavitation."

In connection with the portion of the paper on page 197 reading: "It has been stated that with vessels of types 1 and 3 . . . gradually pass over from curve 1 to curve 2 . . . ," Admiral Dyson said:—

"We had a very illuminating example of that in one of the destroyers which we tried during the war—a Fore River destroyer. The starboard propeller at full speed made between thirty and forty more revolutions per minute than the port propeller. No one could understand what caused the difference. The ship was docked and her struts examined. The two struts had come from different foundries, and on measuring them up the starboard strut was found to be twice as thick as the port strut. That strut was taken out and reduced to the same dimensions as the port strut. At the next trial the propellers ran evenly as to revolutions, there was no loss in efficiency, and we got the same speed with the same power the second time as was obtained the first time."

In connection with the reference to vessel I, Commander Dyson said:—"I was very anxious for Commander McEntee to take the model of this ship and run it in the tank. The engines were designed for something like 7,500 horse-power and were driven up to 9,300 horse-power; the resultant indicated horse-power curves and revolution curves were very peculiar and show the best example of cavitation through a long range

of any ship I have any record of. I was anxious for Commander McEntee, when he was at the tank, to take the model with the model screws, run her through the ranges we have been running through in actual trial, and see if we could not get some cavitation data from the tank, but his time was too fully occupied for him to take it in hand."

At another point:—"In designing propellers for a shipbuilding company you ask your client what minimum speed under full load he expects to make. The answer is: 'That does not make any difference. The guarantee is on the trial conditions.' The screw you need for the light-trial condition is not the screw you need for deep-load conditions. I can give you a screw for trial conditions that will show a very high efficiency, but when you load the ship down and go to sea that screw will go to pieces the first time you strike heavy weather or the ship's bottom gets foul—any adverse conditions at all will knock it to pieces."

At another point:—"Many people have an idea that, because a tugboat has a very full, big-shaped blade, it is just because it is a tugboat. Not so. It is because it must put so much effective horse-power through that screw on a very small diameter, its diameter being limited by the draught of the tug, and it drives you into the broad, fantail blade. You cannot get away from it.

THE PROBLEM OF THE HULL AND ITS SCREW PROPELLER.

By Rear Admiral C. W. Dyson, U. S. Navy, Member.

[Read at the twenty-eighth general meeting of the Society of Naval Architects and Marine Engineers, held in New York, November 11 and 12, 1920.]

1. *Introduction.*—In 1915 Sir Archibald Denny made the statement that "in the future the rules for the correct designing of propellers should be derived from data carefully taken from the trials of smooth bottom vessels carefully run over accurately measured deep water courses."

Several years ago Admiral Taylor wrote to the effect that if theoretical formulas for design are adhered to, it would be necessary to compare each formula with experimental results and select that one which seemed to agree more closely. Then, using this as a semiempirical formula, with constants deduced from experiments or experience, problems could be satisfactorily dealt with.

Both of these statements practically agree. Denny boldly casts all theoretical treatment of the propeller into the discard, while Taylor, by implication, takes the same action, only retaining enough of the theory to save its face, to use a Chinese expression.

To accept these statements of such eminent authorities at their full value throws us at once into the field of empirical results from which to derive data and formulas for the determination of hull forms and characteristics of propellers to fit any particular form of hull and the resistance which it may be desired to overcome.

2. *Difficulties Encountered in Obtaining Reliable Data.*—Upon entering the empirical field for propeller data, the investigator is at first encouraged by the large, apparently large, amount of data available, but it does not require any extravagant amount of time for him to ascertain that fully 95 per cent of the available data is absolutely worthless, this statement of course referring to conditions where general formulas and constants for design covering the entire field of the combined problem, ships and their propellers, are being sought.

The principal causes of worthless data are:—

(*a*) Lack of model-tank trial curves for the ship.

(*b*) Variations in conditions of ship's bottom.

(*c*) Deep sea trials in place of measured mile.

(*d*) Variation in displacement and trim from those of model.

(*e*) Measured mile trials run in a tideway with insufficient number of runs for each spot.

(*f*) Trials run in shallow water.

(*g*) Model hull, when tried, not fitted with external appendages.

All of the above relate to the hull, and the responsibility for them rests with the owner, the builder and the naval architect, all of whom, apparently, are so short-sighted as to be able to see only the few dollars saved to-day by comparatively petty economies, losing sight entirely of the thousands of dollars which would be realized in the future were the small trial savings of to-day ignored.

Turning now to the propellers we find:—

(*h*) Blade forms varying according to the taste of the designing engineer.

(*i*) Blade sections varying to the same degree as blade forms.

(*j*) Variations of pitch in a blade and in the blades of the same propeller.

(*k*) Blades abnormally thick and blunt.

(*l*) Blades exceedingly rough.

(*m*) Blades made with pitch expanding radially or axially, or both.

(*n*) Hub of such form as to cause abnormal eddy losses.

(*o*) Propellers so located with respect to the hull of the ship as to produce abnormal losses.

The guilt for all of these rest with the designer of the propeller. The naval architect is, however, *particeps criminis* in the last-named offense.

3. *Derivation of Fineness of After Body of the Ship.*—In making the selection of data upon which to base formulas and design data, it is necessary to confine consideration to hulls of fine lines, with propellers so placed that the chances of decrease in propulsive efficiency due to hull effect are reduced to a minimum. The propellers which are fitted to these hulls must all have practically the same radial distribution of the projected areas, must have the same general forms of blade sections, must be true to pitch, fine-edged and smooth, in order to eliminate as many variables as possible from the equations.

VARIATIONS IN HULLS OF VESSELS.

Suppose a vessel to have a certain set of forward and of after-body lines with a parallel middle body of a certain percentage of length of the vessel. She will have a certain block coefficient. Now, maintain the same forward and after-body lines but modify the length of the middle body. As the length of the middle body increases over the initial length, the block coefficient increases, and vice versa; but the after-body lines, which control the character and quality of the flow of water to the propeller, have remained as at first. It is impossible, therefore, to base propeller conditions on block coefficient, but there must be provided a means of bringing measurements of block coefficient to a standard condition.

In the foregoing example the constants, in addition to the forward and after lines, are the beam, B, and the draught, H, while the ratio $B \div$ L. L. W. L., or $B \div$ L. B. P., is variable.

Having obtained a means of correcting for the variable, there will immediately be discovered another condition which contains the same variable as before but, in addition, H has become variable. Experience has taught that with constant form and area of load-water plane, to decrease the draught of the hull

is to increase the nominal block coefficient, but the wake will not increase but will remain practically the same as with the initial draught; while should the draught be increased the nominal block coefficient will be decreased but, as in the opposite case, the wake will remain practically unchanged.

Having corrected for varying ratios of length to beam and of draught to beam, it might be considered that the coast had been cleared and that a position had been reached where the propeller could be taken up, but this is not the case. There is still a variable, and that a very important one, to take care of, namely, varying ratios of lengths of after body, or if preferred to so call it, length of run to draught. This variable has a very strong effect on the wake, varying both the character and the quality of the flow of water to the propeller.

All of the varying dimensions now having been covered, it will be found that still another correction must be made, and that is the correction for type of hull. The types of hulls are sharply defined by the character of the after-body lines as the hull fines towards the stern, whether this fining is produced by a rapid dead rise of the bottom, with a very slow decrease in beam at the load-water line, or whether it is produced by a rapid decrease in beam. The latter type can be subdivided into two types, the first having a full and the second a fine midship section, the first tending to concave body section lines and the second to convex. In the first subtype the propellers are to a great extent, or entirely, covered by the submerged hull, while in the second subtype the propeller may be entirely outside the limits of the load-water plane. The first subtype will be spoken of here as "Hulls of Type 1" and those of the second subtype as "Hulls of Type 3."

Where the hull fines rapidly by means of a rapid rise of the bottom towards the stern the vessel is denoted as having a "Hull of Type 2."

As examples of the different types the following may be taken:

Type 1.—U. S. dreadnaughts and destroyers.

Type 2.—Vessels of the ordinary fantail form of stern such as U. S. colliers and the ordinary type of merchant ship, also submarines.

Type 3.—Vessels having the so-called cruiser stern; gunboats Annapolis and Sacramento, battleships of the Virginia class, battle cruisers, scouts.

METHOD OF MAKING HULL CORRECTIONS (FIGS. 1 AND 2, PLATES 71 AND 72).

The determination of after-body form, or, as it may be expressed, the determination of the slip block coefficient of a ship, together with the effect of the variations of ratios of length, beam and draught, can be expressed graphically. With the exception of the correction for ratio of length of after body to draught, which correction is used rather for the correction of apparent slip rather than of slip block coefficient, the preliminary estimate of this coefficient, together with its correction for ratio of draught to beam and of length on load water line to beam, for all the various hull conditions which may be considered as of normal form, are shown on Figs. 1 and 1*a*, Plates 71 and 72.

On Fig. 1, the line X passes through the points whose abscissas are nominal

block coefficients, while the ordinates are the ratios $B \div$ L. L. W. L. The value L. L. W. L. may be replaced by L. B. P. (length between perpendiculars), without any sensible error, as the entire method must be considered as one of approximation.

For the vessels of types 1 and 2, there is given a curve of coefficient of midship section at the top of Fig. 1, for the block coefficients as given by the intersection of the diagonal through the plotted point of nominal block coefficient and the abscissa value 1 with line X. Where the coefficient of midship section for any block coefficient referred to line X is less than the standard for this referred block coefficient as given in Fig. 1, the after-body form approaches the form for type 3.

Correction for $B \div$ L. L. W. L.

Of the standard hulls chosen on which to base the chart, those of type 3 had short middle bodies, which, when eliminated, would have raised the ratio line of $B \div$ L. L. W. L. to $A-B$, while those of types 1 and 2 would have raised this ratio line to $C-D$, the nominal block coefficients decreasing in both cases. Now let it be supposed that the forward and after bodies remain constant but that the length of the middle bodies be varied. As this length increases, the nominal block coefficient will increase until at length the effect of the fineness of the forward and after bodies on the value of the nominal block coefficients will become inappreciable so that the nominal block coefficient as calculated will be unity. On the other hand, the middle body may be entirely eliminated when the nominal block coefficient will be that of the combined fore and after bodies only. These changes of nominal block coefficient will take place approximately along straight lines passing through the plotted point of nominal block coefficient and the point of 1.0 on the scale of abscissas where the ordinate $B \div$ L. L. W. L. is zero. The equation to these lines is—

Nominal block coefficient on

$$X = \frac{B \div \text{L. L. W. L.}}{.28\ (1 - \text{Nom. B. C.}) + (B \div \text{L. L. W. L.})}$$

Now the line X is for S. B. C.'s of standard twin-screw ships of all the types, and the standard B. C. for single-screw ships of type 3 would be that where the line through the plotted position of the actual ship crosses the line W, while for type 2 and type 1 it would be at the point of crossing of the line Z.

Should the vessel have four shafts, for vessels of standard ratios of $H \div B$ and 2 L. A. B. $\div H$, the slip B. C. for the outboard propellers will be the intersection of the correction line with the line X, and, for the inner propellers, the intersection with the line Y. Should the ratio 2 L. A. B. $\div H$ be much greater than the standard, the above corrections will be reversed. As it takes quite a large variation in S. B. C. to make a heavy variation in revolutions, this correction for position may be neglected without serious error and the point on X only be considered.

Correction for Variations in $H \div B$.

The ratios $H \div B$ for type 3 ships, corresponding to the nominal B. C.'s as given by line A—B, are given by the curves on Figs. 1 and 1a, Plates 71 and 72, marked $H \div B$, type 3, standard, while those for types 1 and 2 vessels are given by the curves on the same figures marked $H \div B$, type 2, standard. There will be noted diagonal curved lines extending from above at the left to below at the right of the figures, marked with the values of the nominal B. C.'s as taken from line A—B for type 3, and from C—D for types 1 and 2. These are direction lines to follow in correcting for values of $H \div B$ less than standard, on Fig. 1, and for greater than standard on Fig. 1a.

Example illustrating the above method of estimating the S. B. C. of a vessel.

To illustrate the above, two vessels will be taken:—

Vessel A:	Nominal B. C.	=	.632
	$B \div$ L. L. W. L.	=	.16
	$H \div B$	=	.3
	2 L. A. B. $\div H$	=	14.66

Examination of the after-body lines discloses that these lines fine rapidly due to a rapid falling away in beam and are convex. This indicates that the hull is of type 3 form.

The coefficient of midship section = .930.

Vessel B:	Nominal B. C.	=	.7
	$B \div$ L. L. W. L.	=	.13
	$H \div B$	=	.5
	2 L. A. B. $\div H$	=	13.5
	Coefficient of midship section	=	.98

Examination of the after-body lines discloses that these lines fine rapidly due to a rapid decrease in draught of the actual full body of the hull, the beam being decreased very slowly as the stern is approached, while the screw is completely covered by the hull, in the case of twin screws. These conditions, in conjunction with any value of $H \div B$, indicate a hull of type 2.

Vessel A, Type 3.

Plotting vessel A on Fig. 1 with a nominal B. C. of .632 and a value of $B \div$ L. L. W. L. = .16, and drawing a line through this plotted point and the point on the base line where B. C. = 1.0, this line is found to cross the line A—B at B. C. = .535. Erecting an ordinate from this point on A—B to the value of $H \div B = .3$ below the curve $H \div B$, type 3, standard, and following the direction lines up to this curve, the corrected B. C. on A—B is found to be .505. Through this point .505 on A—B draw another line through the base at 1.0. Where this

line crosses *X*, which is at an abscissa value of .585, will be the corresponding corrected value of the B. C. for wing screws, while where it crosses *W*, at .64, will be the S. B. C. for a single-screw vessel or for the center propeller of a multiple-shaft vessel with an odd number of shafts.

Should the vessel be fitted with wing propellers, a correction should be made for variation of midship-section coefficient from standard, by multiplying the value of the S. B. C., line *X*, by the ratio actual midship-section coefficient ÷ standard coefficient.

Vessel B, Type 2.

Turning now to vessel B, passing the line through its plotted point, nominal B. C. = .7, $B \div L$. L. W. L. = .13, and abscissa value 1.0, and extending the line to *C*—*D*, it will be found to cross this line at the B. C. value of .582. On the curve marked $H \div B$, condition 2, standard, the value of $H \div B$ is found to be .423, therefore the actual value of $H \div B$ being greater than this standard, turn to Fig. 1a and plot the point *H*—*B* = .5, B. C. = .582, above the lower standard *H*—*B* curve, and from this plotted point follow the curves of direction down to this standard line. It will intersect this line at B. C. = .675. Through B. C. = .675 on *C*—*D* and abscissa unity, again draw a line. Where it intersects *X*, at B. C. = .697, will be the S. B. C. for twin propellers, while where it intersects *Z*, at .797, will be the S. B. C. for a single or center propeller vessel.

Vessel C, Type 1.

In vessels of type 1, the determination of S. B. C. is made as for type 2.

With vessels of this type, the character of the wake appears to be nearly similar to that for type 2, while quality is more nearly like that of type 3.

By "character of wake" is meant the degree of obliquity of flow of the water to the propeller disc, while by "quality of wake" is designated the degree of disturbance of the wake and general decrease in its density due to contained air in the water in which the propeller operates.

Conditions of wake are the most favorable with ships of type 1, less so with ships of type 3, and decidedly unfavorable with ships of type 2, although it may be claimed that vessels of this latter type carry much heavier wakes than do the other types and more gain should therefore be realized from the wake than with the other two types. This claim would be a just one were the quality of the wake with the vessels of this type, with which designers ordinarily have to deal, equal to that experienced with vessels of the other types.

It has been attempted in the foregoing to give a slight idea of the difficulties in arriving at the very foundation of the screw-propeller problem before the propeller can even be approached. It is hoped that it has also shown the absurdity of basing any propeller design upon nominal block coefficient without correction. In connection with this the author had the rather amusing experience of being approached by a gentleman with the question as to how he "would determine the slip block coefficient of a vessel to use by looking at her in the drydock." The

same gentleman also made the statement that he was able to determine by inspection the proper diameter and pitch of propeller for any given hull.

It will now be necessary to say a few words concerning the resistance to be overcome by the propeller in driving the hull through the water.

RESISTANCE—HULL AND APPENDAGES.

The question of hull resistances being so fully understood, it is only necessary to state that, for propeller work in developing a system for design purposes, only such vessels should be used as have had their resistances determined by towing the model of the hull in the Model Basin.

It does not suffice to tow the model of the bare hull, but this model should be fitted with all external submerged attachments in order that the true resistance of the hull may be arrived at.

Fig. 2, Plate 73, shows curves of appendage resistance in percentage of bare hull resistance, erected on values of speed of vessel $\div \sqrt{\text{L. L. W. L.}}$, as prepared by the writer from reports of trials of models in the model tank at Washington. These curves are rough approximations, as the actual appendage resistance varies with the form of hull and the amount of power carried by the hull and also with the revolutions at which the propellers are to run, these latter two factors fixing the size of shafting, external lengths of shafting and size of propeller struts.

PROPULSION LOSSES DUE TO TYPE OF HULL AND TO LOCATION OF PROPELLER IN RESPECT TO THE HULL.

The Power Augment Factor "K."

It is a little premature to speak of these losses at this stage of the investigation as they really cannot be studied until the primary design curves have been arrived at, and this result can only be reached by selecting for the early investigations such vessels as, from the high values of the propulsive coefficients obtained, are decided upon as having all such losses reduced to the smallest possible minimum.

The losses, other than those due to poor workmanship in the propeller itself, arise from the reduction in pressure in the area in which the propeller operates, and also from the obliquity of flow of water to the propeller caused by the fullness of lines of the hull interfering with the flow. There may also be serious losses due to eddying around improperly formed stern and rudder posts, around struts of improper section and arrangement, around and between strut bosses and propeller hubs, around the hub itself, and to excessive roughness of surface of the propeller blades and excessive variation of pitch between different blades of the propeller or throughout each blade. Of course these latter named losses are due to poor workmanship and can easily be avoided by care in casting, finishing and setting the propeller blades.

The loss factor, or "power augment," is given the letter "K" to denote it. In previous works of the author this letter "K" has been spoken of as the thrust

deduction factor, but this does not convey all that "K" represents, which, in fact, are the losses which have just been given.

The K factor has the minimum values for hulls of type 1, but these values for this type increase very rapidly as the tip clearance diminishes after a certain limit of tip clearance between the hull and the circumference of the propeller disc has been reached, this distance, so far as the author can determine, being dependent upon the draught of the vessel and the height of this minimum distance above the base line. The formula for obtaining this mean tip clearance is expressed by M. T. C. $= \frac{H + 10H^1}{11}$ where H^1 (wing screws) = (Actual tip clearance $\times$ 10) $\div$ (Actual height of hub center above base line), and H = (actual immersion of upper blade tip in feet $\times$ 14) $\div$ (actual height of tip above base line, in feet.)

Hulls of type 3 stand next in order of loss, while those of type 2 are the heaviest losers, single-screw ships of type 3 having the same loss factor as twin-screw ships of type 2.

Values of K for the different classes of hulls are shown on Fig. 3, Plate 74, the ordinates being mean referred tip clearances for type 1, while the abscissa values are values of K block coefficient, K. B. C., which are obtained from Fig. 1 by taking the intersection of the line through the plotted point Nominal B. C. — $B \div$ L. L. W. L. and the abscissa point of unity, with X, W or Z, according to whether the vessel is a wing or single-screw ship. The B. C. value of this intersection is the K. B. C. for the vessel. The values given by Fig. 3 hold very closely for all vessels of what may be considered normal dimensions. For vessels having extremely great ratios of length of after body to draught, they may possibly decrease to some extent, but there are no data of sufficient accuracy to enable this to be stated definitely.

THE SCREW PROPELLER.

The vessels which are selected to be used must all have propellers having the same radial distribution of projected area but of several different projected area ratios. The vessels selected by the author were of types 1 and 3, but of different B. C.'s and ranged from a S. B. C. of .375 to .60, from a destroyer to a dreadnaught, while the projected area ratios, measured outside of a circle of two-tenths the diameters of the propellers, varied from .6 to .328.

The blades were oval with the widest part of the projection at two-thirds the radius from the center, while the blade peripheries outside the widest part, measured on the arc, were well rounded.

The ships, at trial, were just out of drydock, thoroughly cleaned and painted, and were run over the measured mile in good weather. Each point of the trial curves was obtained by at least three runs at the same or approximately the same revolutions, two runs in one direction and one in the opposite. The highest point of the performance curves was obtained by five runs.

The propellers of all the ships were of manganese bronze, machined to a true

pitch and to the desired blade sections, which were the same type for all the propellers and highly polished.

The trials were run by competent and experienced men, while the data were taken and computed by trained observers and engineers. There was every reason to believe that what were collected were as reliable as could ever be obtained under actual ship conditions.

Derivation of Design Curves.

Taking the derived data and from it obtaining values of pounds indicated thrust per square inch of disc area (I. $T._D$) of the propeller, and corresponding pounds propulsive thrust per square inch of projected area (P. $T._p$), a series of curves were plotted with I. $T._D$ as abscissas and P. $T._p$ as ordinates, and intermediate curves for intermediate projected area ratios were interpolated.

Curves of corresponding tip speeds in feet per minute (T. S.), and of propulsive coefficient (P. C.) were also laid down, as were also the curves of apparent slip (S).

At one point on each of the established curves of I. $T._D$—P. $T._p$ a decided change in conditions was found to occur, the curve either taking a sudden rapid rise or beginning to fall rapidly.

The values of these points in I. $T._D$, T. S., P. C., and S were taken and plotted on values of P. A. ÷ D. A. (projected area ratio), as abscissas and the equations to the curves obtained when possible, that for I. $T._D$ being—

$$\text{I. T.}_D = 28.41 \left(\frac{\text{P. A.}}{\text{D. A.}}\right)^{1.7}$$

while that for T. S. is—

$$\text{T. S.} = 38148 \left(\frac{\text{P. A.}}{\text{D. A.}}\right)^{.8365 + .2 \left(\frac{\text{P. A.}}{\text{D. A.}}\right)}$$

Equations to the curves of P. C. and of S could not be obtained.

These curves are shown on Fig. 4, Plate 75; the conditions covered by them are denoted "Basic Conditions" and are constant with the exception of the S curve which, it will be noted, is based on S. B. C. as abscissas and not on P. A. ÷ D. A. In other words, each standard S. B. C. has its constant standard basic slip.

This curve of S is again shown on Fig. 5, Plate 76, but is there based on values of 2 × length of after body of the ship ÷ the draught, which brings in the final correction for hull form already mentioned.

Correction of S for Varying Values of 2L. A. B. ÷ H.

On Fig. 5 is shown a curve marked "Slips for S. B. C. on X—W, Type 1 and 3 Hulls." This curve is in reality the same as the curve of S shown on Fig. 4, but in the case of Fig. 5, it is erected on abscissa values of 2L. A. B. ÷ H instead of on S. B. C., and the S. B. C. values are marked on it at the standard values of 2L. A. B. ÷ H as determined from Fig. 1.

Experience has demonstrated that as the lengths of after bodies of ships increase or decrease from the standard, the basic values of S decrease or increase and, in decreasing or increasing, follow a general law which may be expressed by the following equation—

$$A(1-S)^a = \frac{2 \text{ L. A. B.}}{H}$$

where A is the length of after body $\div H$ at which the basic apparent slip becomes zero, while the exponent a depends for its value upon A. The lower scale of abscissa is the scale of A upon which the curve of the exponent, a, is erected. The curves developed from the equation for coincident values of A and a are erected on values of 2L. A. B. $\div H$ as abscissas, the scale being that given on the upper scale of abscissas, Fig. 5, while the ordinates are values of basic apparent slip, S.

The curves are direction curves according to which S varies as the ratio 2L. A. B. $\div H$ varies from standard. Thus, suppose the S. B. C. of the vessel be .575. By examination of Fig. 5, it is seen that the curve marked $A = 22.5$ crosses the line of standard S at S. B. C. $= .575$ and 2L. A. B. $\div H = 16.3$. Now suppose the ratio 2L. A. B. $\div H$ for the vessel in question be $17\frac{1}{2}$. The curve $A = 22.5$ is seen to cross the ordinate of $17\frac{1}{2} = 2$L. A. B. $\div H$, at .113, which is the value of S to use in determining the basic condition of the vessel's propellers.

Computation for Basic Condition of Propeller.

To illustrate the method of determining the basic condition of a propeller, let an example be taken, as follows—

Suppose the corrected S. B. C. of the vessel be .575; the ratio 2L. A. B. $\div H$ be 17.5; the propeller blades of standard form and three-bladed.

Let Diameter $= D = 16$ feet

Pitch $= P = 18$ feet

Projected area ratio = P. A. $\div$ D. A. $= .34$.

Then from Fig. 4 we obtain the following data:—

1. Tip speed for P. A. $\div$ D. A. $= .34 = 7{,}480$ feet, from which, since Pitch $\times$ Revolutions $= P \times R = \frac{\text{T. S.} \times P}{\pi D}$.

$$\text{Basic speed} = V = \frac{P \times \text{T. S.}(1-S)}{\pi D \times 101.33}$$

$$= \frac{18 \times 7480 \times .887}{\pi \times 16 \times 101.33} = 23.45 \text{ knots.}$$

2. Indicated thrust per square inch of disc area $=$ I. $T._D = 4.7$ for P. A. $\div$ D. A. $= .34$, then basic indicated horse-power for a three-bladed propeller $=$

$$\text{I. H. P.} = \frac{P \times R \times \text{I. T.}_D \times D^2}{291.8} = \frac{\text{T. S.} \times P \times \text{I. T.}_D \times D}{\pi \times 291.8}$$

$$= \frac{7480 \times 18 \times 4.7 \times 16}{\pi \times 291.8} = 11{,}045$$

and assuming that the shaft horse-power equals 92 per cent of the indicated in well-built and adjusted engines, a figure which experience demonstrates to be nearly correct, basic shaft horse-power = S. H. P. = .92 × I. H. P. = 11,045 × .92 = 10,161.4.

3. The propulsive coefficient for a projected area ratio of .34 is seen, from Fig. 4, to be P. C. = .667, therefore the basic effective horse-power = E. H. P. = I. H. P. × P. C. = 11045 × .667 = 7367.

Variation of the above for four blades and for two blades.

Suppose the propeller should have four blades with a total projected area ratio = .453 or a two-bladed one with a projected area ratio = .227, then—

Blades	4	2
Total P. A. ÷ D. A. =	.453	.227
P. A. ÷ D. A. =	.34	.34

that is, the basic P. A. ÷ D. A. equals ¾ total for the four blades and equals 3/2 total for the two blades.

The values of T. S. and I. $T._D$ are those pertaining to the basic P. A. ÷ D. A., while the propulsive coefficients which are dependent upon the total area ratio will be those corresponding to the total projected area ratios.

The formulas for basic I. H. P. become, for four blades:—

$$\text{I. H. P.} = \frac{\text{T. S.} \times P \times \text{I. T.}_D \times D}{.865 \times \pi \times 291.8} = \frac{7480 \times 18 \times 4.7 \times 16}{\pi \times 252.41} = 12{,}768.$$

for two blades:—

$$\text{I. H. P.} = \frac{\text{T. S.} \times P \times \text{I. T.}_D \times D \times .75}{\pi \times 291.8} = \frac{7480 \times 18 \times 4.7 \times 16}{\pi \times 389.1} = 8{,}283.$$

The basic effective horse-powers now become:—

Four blades; E. H. P. = 12,768 × .585 = 7,470.
Two blades; E. H. P. = 8,283 × .706 = 5,850.

Should the blades be of other than standard distribution of projected area, other corrections occur which the limits of this paper will not allow of discussion.

Conditions Other than Basic (Fig. 6, Plate 77, Curve 4).

Usually the performance of a vessel and its propeller will be very remote from the basic conditions of the propeller. In such cases it is necessary to have a ready means of converting from basic to actual conditions, and such means, so far as power is concerned, are provided as follows:—

Let e. h. p. be the effective horse-power required by a given vessel for a speed v, then—

$\frac{\text{e. h. p.}}{\text{E.H.P.}}$ = fraction of basic load and $\frac{v}{V}$ = fraction of basic speed.

Taking a large number of vessels having hulls of such form and propellers so located that the value of the power augment K will be unity, and from the curves of performance of each vessel obtaining the I. H. P. required to deliver fractional loads, with the speeds v at which these fractional loads were required and denoting this I. H. P. by I. H. P.$_p$ the equation for I. H. P.$_p$ will be:—

$$\text{I. H. P.}_p = \text{I. H. P.} \times \left(\frac{v}{V}\right)^w$$

in which the factor $\left(\frac{v}{V}\right)^w$ is constant for any given value of $\frac{\text{e. h. p.}}{\text{E.H.P.}}$

This equation may be expressed thus:—

$$\text{Log I. H. P.}_p = \text{Log I. H. P.} + w\ (\text{Log } v - \text{Log } V)$$

which, by transposing, becomes—

$$w\ (\text{Log } V - \text{Log } v) = \text{Log I. H. P.} - \text{Log I. H. P.}_p$$

Now represent this value w (Log V — Log v), by the symbol Z_p and, after establishing several values of it at corresponding load fractions, determine the equation to the curve passing through these points and the basic point where $\frac{\text{e. h. p.}}{\text{E.H.P.}}$ and $\frac{v}{V}$ both equal unity.

This equation will be—

$$Z_p = 1.0414 - 1.0414 \text{ Log } \left(\frac{10 \text{ e. h. p.}}{\text{E. H. P.}}\right)$$

or, to simplify it,

$$Z_p = 1.0414 \text{ Log } \left(\frac{\text{E. H. P.}}{\text{e. h. p.}}\right)$$

for values of $\frac{\text{e. h. p.}}{\text{E.H.P.}}$ less than unity, and

$$Z_p = 1.0414 \text{ Log } \left(\frac{\text{e. h. p.}}{\text{E. H. P.}}\right)$$

for values of $\frac{\text{e. h. p.}}{\text{E.H.P.}}$ greater than unity; these values are subtractive from Log I. H. P. in the first case and additive in the second, so that for values of

$$\frac{\text{e. h. p.}}{\text{E.H.P.}} < 1, \text{ Log I. H. P.}_p = \text{Log I. H. P.} - Z_p$$

$$\frac{\text{e. h. p.}}{\text{E.H.P.}} > 1, \text{ Log I. H. P.}_p = \text{Log I. H. P.} + Z_p.$$

Should the efficiency of the combined hull and propeller be such that the value

of the power augment factor "K" is greater than unity, then denoting the total power by I. H. P.$_d$,

$$\text{I. H. P.}_d = K \times \text{I. H. P.}_p = \frac{K \times \text{I. H. P.}}{10^{Z_p}}, \text{ or}$$

$$\text{Log I. H. P.}_d = \text{Log I. H. P.} + \text{Log } K \pm Z_p.$$

These equations neglect all correction for the effects of cavitation which will be discussed further on.

Problems Illustrating Analysis of Propeller.

In order to illustrate all that has been treated in the foregoing, two problems are here given, the first being the series of vessels used by Commander McEntee in his article on the "Propulsive Efficiency of Single-Screw Cargo Ships" which was read before this Society at its 1919 annual meeting.

It must be remembered, however, that in Commander McEntee's vessels the ships were not evenly balanced in their lines, the effect of shifting the middle body aft having its greatest effect on that vessel having the resultant shortest after body. As the curves on Fig. 1 for estimation of S. B. C. are based on forward and after bodies of equal length, the result may be a slight under estimate of S. B. C. and of K in the case of one or more of the vessels of the fuller nominal block coefficients.

Problem 1: Vessels of Type 2, Single Screw.

L. B. P. = 400′	$\frac{2 \text{ L. A. B.}}{\text{H}}$ =	11′.9	11′.4	10′.88	10′.37	9′.86
B = 57′.3						
H = 26′	Nom. B. C.	.7242	.7438	.7634	.7830	.8026
$\frac{\text{B}}{\text{L. B. P.}}$ = .1432	K. B. C.	.758	.773	.786	.801	.815
	K	1.175	1.2	1.23	1.27	1.31
$\frac{\text{H}}{\text{B}}$ = .454	S. B. C.	.7925	.813	.825	.84	.855
	S	.14	.155	.17	.195	.2175
D = 16′.583						
P = 14′.75						
$\frac{\text{P. A.}}{\text{D. A.}}$ = .267						
Blades = 3						
T. S. = 6230						
P × R = 1764						
1 − S = .86	.845	.83	.805	.7825		
V = 14.97	14.71	14.45	14.01	13.8		
I. T.$_D$ = 3.09						
I. H. P. = 5137						
P. C. = .696						
E. H. P. = 3575						
S. H. P. = 4726						

v=11	11	11	11	11
$v \div V$=.7348	.7479	.7614	.785	.8076
e. h. p.=1160	1200	1260	1370	1580
e. h. p.÷E. H. P.=.3244	.3357	.3525	.3832	.442
Z_p (Fig. 6)=−.51	−.492	−.47	−.432	−.382
K=1.175	1.2	1.23	1.27	1.31
S. H. P.$_d$=1716	1827	1970	2220	2569
Act. Power=1750	1850	1980	2200	2700

By comparison of the estimated and actual powers it will be seen that the only difference of serious amount is with the vessel having the heaviest nominal B. C., as was to be expected due to the excess fullness of the after body lines over those of an equally balanced ship of the same nominal B. C. This difference existing is, however, too great to be credited entirely to this cause.

There is another explanation for the excess of actual power over estimated in the case of the fifth vessel, and this is, in all probability, the correct one. It will be noted that the value of K in this case is 1.31 while that of Z_p is —.382. Taking .382—Log K as the value of Z_p corresponding to the total work (load fraction) being delivered by the propeller, the value $\frac{\text{e. h. p.}}{\text{E.H.P.}}$ corresponding to this is found from Fig. 6, to be .555.

Plotting this point on Fig. 7, Plate 78, with $v \div V = .8076$ as ordinate, it will fall below the curve of $S = .2175$ for vessels of type 2, slips of the second order. Now assume that the existence of a K loss produces earlier cavitation and that the propeller in the case in question is cavitating. The value of $\frac{\text{e. h. p.}}{\text{E.H.P.}}$ corresponding to the point on the curve $S = .2175$ whose ordinate is $\frac{v}{V} = .8076$, is approximately .525, and the value of Z_p corresponding to this is .29, then the increase log factor due to cavitation is .29 —(.382 —.11727) = .02527 and the corrected value of I. H. P.$_d$, taking cavitation into account, will be:—

$$\text{Log I. H. P.}_{dc} = \text{Log I. H. P.}_d + .02527 = 3.43503$$

I. H. P.$_{dc}$ = 2723, which corresponds very closely with the actual power, 2700.

Now K undoubtedly does produce earlier cavitation; therefore, when deciding upon the design point on Fig. 7, it should be so located that the $\frac{\text{e. h. p.}}{\text{E.H.P.}}$ point, corrected for K, will fall well above the basic curve of S for the vessel. The remainder of the vessels in the McEntee problem plot above their curve of S when the correction for K is made, and therefore are free from any cavitation.

Assuming that the above is correct, as it undoubtedly is, the supposition that model tank trials give no indication of cavitation is incorrect, as in the case just discussed the model propeller is clearly indicating "Dispersal of the Thrust Column," a term which will be explained later.

Problem 2: Vessel of Type 3.

L. L. W. L. = *
B = *
H = *
$\frac{H}{B}$ = .269
$\frac{B}{\text{L. L. W. L.}}$ = .124
$\frac{2\text{L. A. B.}}{H}$ = 30.09
Nom. B. C. = .577
B. C. on $A-B$ = .418
B. C. on $A-B$ cor. for $\frac{H}{B}$ = .34
Slip B. C. on X = .45; S = .0775
K. B. C. = .51
K = 1
Number of propellers = 4
D = *
P = *
$\frac{\text{P. A.}}{\text{D. A.}}$ = *

Blades = 3
Form = Standard
T. S. = *
$P \times R$ = 5772
$1-S$ = .9225
V = 52.55
I. T.$_D$ = *
I. H. P. = 225740
P. C. = .525
E. H. P. = 118510
S. H. P. = 207680
v = 32
$\frac{v}{V}$ = .609
e. h. p. = *(bare hull) = *(est. all append.)
$\frac{\text{e. h. p.}}{\text{E.H.P.}}$ = *
Z_p = *
K = 1
S. H. P.$_d$ = 155,380
Act. power = Between 150,000 and 155,000

Having illustrated the method followed in estimating the power required to drive a vessel at any given speed against the resistance existing at that speed, it remains to explain the method of estimating the revolutions at which the propeller will turn under these same conditions of estimated power and speed, and this will be taken care of now under the subhead—

Estimate of Revolutions under Other than Basic Conditions of Operation.

Commander S. M. Robinson, whom the author had the pleasure of calling his assistant during the period of the Great War, by laying down curves of revolutions of propellers of standard form with actual speeds through the water as abscissas, derived a curve of exponents, y, for which the equation is as follows:—

$$y = \frac{v}{v-1}\left(2.626 + \frac{5861.3}{28045 + (v-25)^4}\right)$$

A curve of Log y was also erected on v as abscissas and the values of the logs of basic V^y and v^y were denominated Log A_V and Log A_v.

The equation for apparent slip at any speed was found to be

$$s = S\frac{\text{I.H.P.}_d \times V^y}{\text{I.H.P.} \times v^y} = S\frac{KV^y}{10^{Z_p}v^y}, \text{ or}$$

*The data omitted are confidential and therefore cannot be made public.

$$\text{Log } s = \text{Log } S + \text{Log I.H.P.}_d + \text{Log } A_V - \text{Log I.H.P.} - \text{Log } A_v$$
$$\text{Log } s = \text{Log } S - Z_p + \text{Log } A_V - \text{Log } A_v \text{ when } K = 1.$$

Now let $\text{Log } A_V - \text{Log } A_v = Z_S$, and plotting it on values of $\frac{v}{V}$ as shown by curve 1 on Fig. 6, it will be found to be expressed by the equation—

$$Z_S = 2.861 \text{ Log } \left(\frac{V}{v}\right) - .0788.$$

This curve reaches the value $Z_S = 0$ at a value of $\frac{v}{V} = .9385$ and this value of 0 is retained at least until $\frac{v}{V} = 1$ is reached. Beyond that the question of cavitation enters and certain variations may occur, depending upon the values of $\frac{v}{V}$ and the corresponding values of $\frac{\text{e. h. p.}}{\text{E.H.P.}}$

The form of the equation for apparent slip, where $K = 1$, now becomes

$$\text{Log } s = \text{Log } S - Z_p + Z_S \text{ and when } Z_p = Z_S$$

$s = S$, that is, when the vessel is loaded down so that at any fraction of load $\frac{\text{e. h. p.}}{\text{E.H.P.}}$, and corresponding speed v, when Z_S for $\frac{v}{V} = Z_p$ for $\frac{\text{e. h. p.}}{\text{E.H.P.}}$, the apparent slip has become equal to the basic slip and the propellers are passing into the cavitating zone, with or without loss, depending upon the quality of the wake which the vessel carries.

To illustrate the method of estimating revolutions when cavitation does not exist, the calculations for Problems 1 and 2 will be continued.

Estimate of Revolutions.

Problem 1. $v = 11$

S. H. P.$_d$	1716	1827	1970	2220	2569
$\frac{v}{V}$	.7348	.7479	.7614	.785	.8076
Z_S for $\frac{v}{V}$ (Fig. 6—1)	+.298	+.280	+.25	+.217	+.182
s	.1010	.1142	.126	.1510	.1798
$R_d = \frac{v \times 101.33}{P \times (1-s)}$	84.07	85.31	86.46	89	92.14
Actual revolutions	84.5	85.5	86	88	93.2

Problem 2. $v = 32$.

S. H. P.$_d$	155,380	s	.1965
$\frac{v}{V}$	.609	R_d	209.6
Z_S	+.53	Actual revolutions	208+

Before leaving the subjects of estimation of power and of revolutions it should be of interest to note that none of the vessels covered by the above problems was used to any extent whatever, in the derivation of the curves of design and estimating.

The processes of estimating under normal conditions having been explained, it is now necessary to take up the same subject under abnormal conditions of load, which abnormal conditions are treated under the general head of—

Cavitation (*Fig. 7, Plate 78*).

It has been shown how vessels can be divided into three general types according to the form of the after-body lines. They may also be divided into types according to the values of the basic slips, namely:—

1. Vessels having basic slips of the first order, which include vessels having basic slips of about 13 per cent and lower.

2. Vessels having basic slips of the second order, which include all vessels having basic slips exceeding about 13 per cent.

These distinctions according to the basic slips are encountered with vessels of type 2, and, in order to explain them, attention is called to Figs. 6 and 7.

Many years ago, on the trials of the British destroyer Daring, difficulty was experienced in obtaining the contract speed, due to failure of the propellers, such failure being accompanied by heavy vibration. The difficulty was finally overcome by increasing the projected area of the propellers to quite a large extent. After a thorough study of the phenomenon, Mr. S. W. Barnaby, of Thornycrofts, advanced the theory that the failure was due to a lack of flow of water to the propeller caused by insufficient supply head, this failure resulting in the formation of cavities in the water in which the propeller was working. He gave the name of "Cavitation" to this action and fixed a limit of effective thrust, 11½ pounds per square inch of projected area, as that at which cavitation could be expected to occur. Later experience has demonstrated that this limit is not correct but that it depends upon the projected area ratio; the higher this ratio the higher the effective thrust before cavitation happens. Taylor supposes that both tip speed of the propeller and effective thrust enter into the solution of the problem. By inspection of Fig. 4, curves of design, it will be noted that tip speed increases with indicated thrust and with effective thrust as the projected area ratio increases. It is also a fact that, in changing from the basic conditions of resistance, for equal powers the indicated thrusts vary inversely as the tip speeds. Therefore, with a vessel loaded down very heavily beyond the designed condition the tip speeds will, as a rule, be less than the designed tip speeds and yet, as experience has taught, "cavitation" may occur.

It is not the intention of this paper to advance any theories as to cavitation, but rather to have the student form his own theories after having had pointed out to him the various phenomena that actual performance data have shown to occur.

For convenience in discussion, the phenomena encountered where the load

fraction $\frac{\text{e. h. p.}}{\text{E.H.P.}}$ is less than unity will be spoken of as "cavitation of" or "dispersal of the thrust column" of the propeller, while for those phenomena met with where the load fraction exceeds unity the term "cavitation of the suction column" of the propeller will be employed.

Dispersal of the Thrust Column.

It has already been stated that vessels can be divided into types according to the magnitude of their basic slips and that vessels of type 2 exhibit the variations due to changes in this magnitude.

Turn now to Fig. 6. Without cavitation, it has already been pointed out that to obtain the value of the apparent slip for any other than basic conditions the equation is:—

$$s = S\frac{\text{I.H.P.}_d \times 10^{Z_s}}{\text{I. H. P.}}$$

or Log s = Log S + Log K + $Z_S - Z_p$, where Z_S is taken from curve 1.

With hulls of type 2, at any given value of load fraction there will be a corresponding value of speed fraction $\frac{v}{V}$, at which the quality of the wake will change materially; in the cases of vessels of type 2, having basic apparent slips of the first order, two changes will occur and there will be two corresponding values of $v \div V$. The first change with such vessels will be from a basic slip of the first order to one of the second order, while the second change will be from slips of the second order to cavitating slips. The first change will occur without loss of efficiency of propulsion when the value of K equals unity. When the value of K is greater than unity, the change of slips will be accompanied by an increase in value of K and all the phenomena of cavitation will occur. When K equals unity the second change will be accompanied by a rapid loss in efficiency of propulsion.

When the first change occurs, if it were known what the new basic S were to be, it could be inserted in the equation for apparent slip in place of the original basic S, and the values of Z_S be taken from curve 1, Fig. 6, as before. In place of making this change of basic slip, available data have given the curve marked "2" from which, using the value of the original basic S, the new values of Z_S are taken. For this curve of Z_S, the values of Z_S vary as 3.09 Log $\left(\frac{V}{v}\right)$.

Now assuming that when the actual apparent slip at any gross load fraction has reached the value of the basic S, apparent slip conditions are on the verge of change,

$$\text{Log } s = \text{Log } S + Z_S - Z_p + \text{Log } K$$
$$s = S,$$
$$\therefore Z_S + \text{Log } K = Z_p.$$

Turn now to Fig. 7. The abscissas of this figure are values of load fraction $\frac{\text{e. h. p.}}{\text{E.H.P.}}$, while the ordinates are values of speed fraction $\frac{v}{V}$. The point whose abscissa value and ordinate value are both unity is the point of basic condition, while diagonals passing through the origin and crossing the ordinate $\frac{v}{V} = 1$, at any value of $\frac{\text{e. h. p.}}{\text{E.H.P.}}$ are diagonals of effective thrust ratios, the ratio being equal to the load fraction $\frac{\text{e. h. p.}}{\text{E.H.P.}}$ at which the diagonal intersects the ordinate $\frac{v}{V} = 1$, and is expressed by $\frac{\text{e. t.}}{\text{E.T.}}$, in which e. t. is the actual effective thrust while E. T. is the basic thrust of the propeller.

Lay down the curve where $Z_S = Z_p$, one of the points of which must be the point of basic condition, and it will be found that the curve, marked A, will follow the law—

I. H. P.$_p$ or S. H. P.$_p$ varies as 3.09 Log $\left(\frac{v}{V}\right)$ or Z_p varies as 3.09 Log $\left(\frac{V}{v}\right)$, which is the same as that for Z_S when $s = S$.

Calling the new basic slip of the second order, S_1, the ordinate of any point on A, $\frac{v}{V_1}$; the value of Z_S for $\frac{v}{V_1}$ taken from curve 1, Fig. 6, Z_{S1}; the original basic slip, S, the corresponding value of the ordinate $\frac{v}{V}$; and the value of Z_S for $\frac{v}{V}$, taken from curve 2, Fig. 6, Z_S,

$$\text{Log } S + Z_S = \text{Log } S_1 + Z_{S1},$$

from which for any assumed values of S_1, $\frac{v}{V_1}$ with its corresponding value of Z_{S1}, can be found a curve of values of S for a series of assumed values of $\frac{v}{V}$ (each less than $\frac{v}{V_1}$), with the corresponding values of Z_S. Taking a series of values of S_1 for any value of $\frac{v}{V_1}$, and it will be found that as S_1 increases S will increase and $\frac{v}{V}$ diminish until S_1 equals approximately 21 per cent, when the value of S will become constant for any further decrease of $\frac{v}{V}$.

The values of $\frac{v}{V}$ and the values of S at which the latter becomes constant are the limits at which the change from basic slips of the first order to those of the second order occurs, and this statement is borne out by all the data available. The curve of these slips is shown at the lower part of Fig. 7, while the limiting curve of $\frac{v}{V}$ is shown, marked B. For type 1 and type 3 vessels, this shift of Z_S values does not occur, so far as can be ascertained, but Z_S is taken always from curve 1, Fig. 6.

These two types and type 2 of first order of apparent slips having $K=1$, pass on down to $\frac{v}{V}$ values shown by curves C, Fig. 7, before genuine "Dispersal of the Thrust Column," with consequent loss in efficiency, is encountered. These curves, C, have been determined from the performances of a four-shaft vessel when steaming with only one propeller in use on each side of the ship, the idle propellers dragging, a slight correction having been allowed in the e. h. p. curve of the vessel for the increase in resistance due to these idle screws. The actual increase occurring is one that should be solved by the Model Tank.

Turning now to hulls of type 2 with basic apparent slips of the second order, only one change occurs in the slip conditions, and that is the change from slips of the second order to those of cavitation. These latter are augmented (similar to those of types 1, 3, and of 2 when original slips are of the first order, when $\frac{v}{V}$ is less than those of curves C (Fig. 7), on entering cavitation due to the increase in power demanded by the cavitating condition.

As the curve B is the change limit for slips given by the slip curve D, it is assumed that a corresponding change from slips of the second order to those of cavitation will occur at values of $\frac{v}{V}$ varying inversely as the ratios $\frac{1-S \text{ of } B \text{ curve}}{1-S \text{ of 2d order}}$, and by assuming different values of these slips of the second order, curves of cavitation limits varying from $S=.27$ down to curve B have been laid down.

For hulls of type 2, curve B has been checked by the performances of submarines of the Holland type, while the curves for slips of the second order have been checked by submarines of the Lake type. These two types usually are radically different in the forms of hulls and in the arrangement of their propellers. Vessels of the Holland type require the use of the surface draught in obtaining the S. B. C., while only one-half diameter of the hull is used in determining the ratio $2\text{L.A.B.} \div H$, and those of the Lake boats, while obtaining the S. B. C. in the same manner as for the former, use a much greater value than one-half diameter of hull in obtaining the after-body length ratio; in fact, for both types the H in the ratio $2\text{L. A. B.} \div H$ equals the height of the point of the stern above the base line.

This results in a much greater value for the after-body ratio of the Hollands than for the Lakes, with a corresponding reduction in the basic slip values. Due to the position of the propellers in relation to the hull and also to the peculiar form of hull, the Lake boats have a shifting value of K depending upon the length of the vessels until this length equals or exceeds 230 feet, when K becomes unity for all surface conditions of speed. For all lengths of vessels of the Holland type operating on the surface, K has a constant value of unity.

When vessels of either type are running submerged it becomes necessary to bring the diving rudders into operation in order to maintain proper balance, and the additional resistances introduced by the resistances of the diving rudders appear to vary with the speeds, disappearing entirely at 10 knots with boats of the

Lake type but reappearing at speeds above 10 knots. With vessels of the Holland type, the resistance due to the influence of the diving rudders appears to reach a maximum at 7 knots, remaining practically constant in its effect on K until 9 knots is reached, when it begins to fall rapidly.

In both types of vessels the value K at any speed appears to depend upon the length of boat; in the case of the Holland boats, K in the submerged condition becomes unity when a load-water-line length of 200 feet has been reached, while with the Lake type the value of unity is not reached until a load-water length of 230 feet has been attained.

These values of K for both types of vessels are given in the following tables:—

VALUES OF K FOR SUBMARINES.

Lake—Submerged.

L.W.L.	6	7	8	9	10	11	12
	K.						
150	1.35	1.24	1.14	1.05	1.0	1.1	1.4
160	1.30	1.20	1.12	1.04	1.0	1.09	1.34
170	1.25	1.17	1.10	1.04	1.0	1.07	1.28
180	1.20	1.13	1.08	1.03	1.0	1.06	1.22
190	1.14	1.10	1.06	1.02	1.0	1.04	1.16
200	1.09	1.06	1.04	1.015	1.0	1.025	1.11
210	1.05	1.035	1.02	1.01	1.0	1.02	1.05
220	1.00	1.00	1.00	1.00	1.0	1.0	1.0
230	1	1	1	1	1	1	1
250	All below = unity up to 12 knots.						
260							
270							
280							
290							
300							

Values of K for Submarines.

Lake—Surface.

V

L.W.L.	6	7	8	9	10	11	12	13	14	15	16	17
	K											
150	1.17	1.13	1.085	1.04	1.00	1.027	1.095	1.14	1.19	1.23	1.29	1.45
160	1.145	1.12	1.075	1.038	1.00	1.023	1.083	1.112	1.17	1.21	1.25	1.35
170	1.125	1.097	1.064	1.033	1.00	1.02	1.073	1.105	1.143	1.175	1.215	1.3
180	1.105	1.082	1.055	1.03	1.00	1.019	1.062	1.089	1.12	1.147	1.18	1.25
190	1.086	1.068	1.045	1.025	1.00	1.017	1.05	1.072	1.097	1.12	1.148	1.2
200	1.067	1.052	1.036	1.02	1.00	1.013	1.04	1.055	1.075	1.09	1.11	1.15
210	1.047	1.037	1.025	1.015	1.00	1.01	1.03	1.04	1.052	1.063	1.075	1.1
220	1.025	1.020	1.013	1.01	1.00	1.01	1.015	1.02	1.03	1.035	1.04	1.05
230	1	1	1	1	1	1	1	1	1	1	1	1
250	All below = unity until $\frac{v}{\sqrt{L}} = 1.12$, when there may possibly occur a rapid increase in K, reach-											
270	ing 1.22 at $\frac{v}{\sqrt{L}} = 1.19$ and rising rapidly.											
270												
280												
290												
300												

Holland Type.

Submerged only as $K = 1$ for Surface.

v	5	6	7	8	9	10	10.5	11
L.W.L.	K							
150	1.275	1.385	1.395	1.395	1.395	1.375	1.34	1.25
160	1.12	1.23	1.25	1.25	1.25	1.21	1.19	1.07
170	1.04	1.12	1.14	1.14	1.14	1.1	1.09	1.0
180	1.0	1.04	1.06	1.06	1.06	1.03	1.025	1.0
190	1.0	1.005	1.01	1.00	1.00	1.00	1.00	1.00
200	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00
	All greater lengths of L. W. L. have $K = 1$.							

These K's are not true power augments but are genuine variable changes in resistance, and therefore are not treated as the K in Problem 1, in the determination of cavitation.

To illustrate the changes occurring with type 2 vessels with true $K=1$, in passing, first from slips of first order to slips of the second order, and second from slips of the second order to cavitation, analyses of the performances of two vessels are given. The first vessel used is a Holland submarine having a basic slip of the first order, the problem being divided into two parts:—(*a*) Surface condition; (*b*) submerged condition:

VESSEL C.

Hull.

L.L.W.L. $=268.57'$
B $=22.16'$
H $=14.16$ (surface)
$H_1=\frac{B}{2}=11.08'$

Nom. B.C. $=.459$

$\frac{B}{\text{L.L.W.L.}}=.0825$

$\frac{H}{B}=.684$

L.A.B. $=\frac{\text{L.L.W.L.}}{2}$

$\frac{2\text{L.A.B.}}{H_1}=24.24$

S.B.C. $=.451$
S $=.125$
K $=1$

Propeller.

D . . . . . $7'$
P . . . . . $6.25'$
Blades 3
Projec. form . . . . . Standard

$\frac{\text{P.A}}{\text{D.A.}}$ 4

T.S. 8520

$1-S$ 875

V 20.91

I.T._D 6

I.H.P. . . . . 4879 (two propellers)
P.C. 62
E.H.P 3025
S.H.P. 4489

Surface Performance.

e. h. p. / E.H.P.	e. h. p.	v	$\frac{v}{\sqrt{V}}$	Z_p	K	S.H.P.$_d$	Act. S.H.P.$_d$	Z_S, curve 1, Fig. 6	S	s	Revs.	
											Est.	Actual
.05	152	8.44	.404	−1.355	1	198		+1.043	.125	.0609	145.7	145
.075	228	9.57	.458	−1.1715	1	302		+.89	.125	.0654	166	166
.10	304	10.5	.502	−1.0414	1	408		+.77	.125	.0669	182.5	182.5
.15	456	11.86	.567	− .858	1	623		+.62	.125	.0723	207.3	207.5
.20	608	13.05	.624	− .7279	1	840		+.497	.125	.0735	228.4	228
.30	912	14.87	.711	− .5445	1	1281		+.335	.125	.0772	261.3	258.5
.40	1216	16.14	.772	− .4144	1	1729		+.233	.125	.0823	285.2	281.5
.50	1520	17.3	.827	− .3135	1	2181		+.154	.125	.0866	307.1	303.2
.60	1824	18.5	.885	− .231	1	2637		+.070	.125	.0863	328.3	328
.70	2128	19.36	.926	− .1613	1	3096		+.019	.125	.0901	345	347
.80	2432	20	.957	− .1009	1	3558		+.0	.125	.0991	359.9	363
.90	2736	20.5	.980	− .0477	1	4022		+.0	.125	.112	374.3	375

Submerged Condition.

e. h. p. / E.H.P.	e. h. p.	v	$\frac{v}{\sqrt{V}}$	Z_p	K	S.H.P.$_d$	Act. S.H.P.$_d$	Z_S, curve 2, Fig. 6	S	s	Revs.	
											Est.	Actual
.025	76	5.05	.242	−1.667	1	96.7	105	+1.84	.125	.1862	100.6	102
.05	152	6.44	.308	−1.355	1	198.2	109	+1.575	.125	.2075	131.8	129
.075	228	7.33	.351	−1.1715	1	302.5	280	+1.4	.125	.2116	150.7	146
.10	304	8.06	.386	−1.0414	1	408	364	+1.275	.125	.2141	166.3	161
.15	456	9.26	.443	− .858	1	622.5	550	+1.09	.125	.2133	190.8	184
.20	608	10.26	.491	− .7279	1	840	750	+ .95	.125	.2085	210.2	203
.30	912	11.85	.567	− .5445	1	1281	1170	+ .76	.125	.2053	241.8	235.5
.40	1216	13.15	.629	− .4144	1	1729	1620	+ .622	.125	.2016	267.1	262

Unfortunately, in the case of this vessel no measure of shaft horse-power on the surface was taken, the vessel being driven by Diesel engines. In the submerged condition electric motors were the propelling units. Attention is called to the extremely close agreement between estimated and actual revolutions, the slips not, as yet, having changed their character from those of the first to those of the second order. In the submerged condition the actual powers were obtained by measurements of electric input and voltage to the motors and the resulting K. W.

transformed to S. H. P. by dividing by the factor .746. The estimated S. H. $P._d$ is seen to be somewhat in excess of the actual, and this excess estimate is accompanied by an excess in estimated revolutions over the actual. Should the actual measured power at the highest submerged speed be used in the calculation of revolutions for that speed, the resulting estimated revolutions become 264.1 instead of 267.1.

The vessel just discussed has an after body of such form that it is almost at the dividing line between vessels of type 2 and type 1, and the character and quality of the wake are so good that there is no augment of power, that is, $K=1$.

Should the vessel be one of type 2, slips of the first order, but with the character and quality of the wake such that K is in excess of unity, the performance becomes radically different. Upon entering cavitation the value of K becomes increased to approximately $K_c=1.26\,K$.

Suppose in the case of such a vessel a series of performances be taken in which the net load fraction $\frac{\text{e. h. p.}}{\text{E.H.P.}}$ is constant but the speed fraction $\frac{v}{V}$ for this load fraction is decreasing due to progressive loading of the vessel, then

$$\text{Log I. H. P.}_d=\text{Log I. H. P.}+\text{Log } K-Z_p.$$

Now let

$$Z_{pg}=Z_p-\text{Log } K,$$

that is Z_{pg} is the gross load factor for the propeller when delivering the net load fraction $\frac{\text{e. h. p.}}{\text{E.H.P.}}$, with a power augment K. Call the gross load fraction $\frac{\text{e. h. p.}_g}{\text{E.H.P.}}$.

When $\frac{v}{V}$ has decreased until it reaches the intersection of the ordinate at $\frac{\text{e. h. p.}_g}{\text{E.H.P.}}$ with the cavitating curve of the basic slip S, of the first order, cavitation ensues and K changes value to $K_c=1.26\,K$.

The equation for power then becomes

$$\text{Log I. H. P.}_d=\text{Log I. H. P.}+\text{Log } K_c-Z_p.$$

As the value of $\frac{v}{V}$ still decreases, call the value of the load factor corresponding to the point on the cavitation curve of S whose ordinate is $\frac{v}{V}$, Z_1, the equation for power becomes

$$\text{Log I. H. P.}_d=\text{Log I. H. P.}+\text{Log } K_c-Z_p+Z_1-Z_{pg}$$

but $Z_p-\text{Log } K_c=Z_{pg}$, therefore

$$\text{Log I. H. P.}_d=\text{Log I. H. P.}-2Z_{pg}+Z_1.$$

As in Problem A, the slip before cavitation equals $s = S\frac{\text{I. H. P.}_d\ 10^{Z_S}}{\text{I. H. P.}}$, where Z_S is taken from curve 1, Fig. 6, but as soon as the cavitation curve of S is passed, while the equation remains the same, the value of Z_S must be taken from curve 2, Fig. 6.

Turning now to the second vessel, which is of the Lake type, the range of apparent slips of the second order is entered. In the estimates of revolutions both in and out of cavitation, the values of Z_S are taken from curve 1, Fig. 6, but in the estimates of power in the cavitation range a corrective factor must be introduced. Let this factor be called K_1. Its values are found as follows:

Turning to Fig. 7, attention is called to two points marked Z_p, one plotted on $\frac{\text{e. h. p.}}{\text{E. H. P.}} = .575$ and $\frac{v}{V} = .78$, while the other is on $\frac{\text{e. h. p.}}{\text{E. H. P.}} = .5$ and $\frac{v}{V} = .51$. Now for either of these points to find the value of K_1, take the value of $\frac{\text{e. h. p.}}{\text{E. H. P.}}$, called $\frac{\text{e. h. p.}_1}{\text{E. H. P.}}$, where this same value of $\frac{v}{V}$ cuts the cavitating thrust curve for the basic slip S of the vessel. The Z_p value for this value of $\frac{\text{e. h. p.}_1}{\text{E. H. P.}}$ call Z_1, then $K_1 = Z_1 - Z_p$ and

$$\text{Log I. H. P.}_d = \text{Log I. H. P.} + \text{Log } K - 2Z_p + Z_1$$
$$\text{Log S. H. P.}_d = \text{Log S. H. P.} + \text{Log } K - 2Z_p + Z_1$$

and, by transposing the equation,

$$Z_1 = \text{Log S. H. P.}_d + 2Z_p - \text{Log S. H. P.} - \text{Log } K$$

is obtained from which, when the actual power and load fraction are known, the cavitation loss can be closely approximated.

Also, by transposing, the equation—

$$2Z_p = \text{Log S. H. P.} + \text{Log } K + Z_1 - \text{Log S. H. P.}_d,$$

from which the load fraction $\frac{\text{e. h. p.}}{\text{E. H. P.}}$ corresponding to Z_p can be ascertained.

Vessel D.

Hull.	*Propeller.*	
L. L. W. L. = 203.75′	Blades	3
$B = 19'$	Shafts	2
$H = 16.1'$	Form	Standard
$H_1 = 15.1'$	D	6.17′
$\frac{B}{\text{L. L. W. L.}} = .093$	P	5.542′
$\frac{H}{B} = .847$	$\frac{\text{P. A.}}{\text{D. A.}}$	.4

L. A. B. = ½ L. L. W. L.		T. S.	8,530
2 L. A. B. ÷ H_1 = 13.5		1 − S	.79
Nom. B. C. = .481		V	18.65
B. C. on C—D = .35		I. T.$_D$	6
B. C. on $C-D$ corrected for $\frac{H}{B}$ = .52		I. H. P.	3,830
		P. C.	.62
S. B. C. on X = .55		E. H. P.	2,371
S = .225		S. H. P.	3,523

Surface Condition—Analysis.

v	e. h. p.	$\frac{\text{e. h. p.}}{\text{E. H. P.}}$	Z_p	K	S. H. P.$_d$		$\frac{v}{V}$	Z_S	S	s	Revs.	
					Est.	Act.					Est.	Act.
6	60	.0253	1.667	1.06	80.4	80	.3217	1.32	.225	.1073	122.9	.....
8	144	.05075	1.32	1.042	176	200	.4289	.975	.225	.1061	163.6	160.1
10	268	.1131	.98	1.0	369	370	.5361	.688	.225	.1149	206.6	203.4
12	444	.1873	.757	1.037	639	632	.6433	.46	.225	.1177	248.7	247
14	740	.3122	.522	1.068	1131	1134	.7506	.2784	.225	.1371	296.7	300.5
16	1050	.443	.362	1.10	1684	1670	.8578	.11	.225	.1386	339.6	345
17	1360	.574	.25	1.13	2239	2500?	.9114	.035	.225	.155	367.9	362.5

The power data for the 17-knot point are doubtful as there was no way of measuring it, and the shop brake-test power for the revolutions was used.

Submerged Condition.

v	e.h.p.	$\frac{\text{e. h. p.}}{\text{E. H. P.}}$	Z_p	$\frac{v}{V}$	$\frac{\text{e.h. p.}_1}{\text{E.H.P.}}$	Z_1	K	S. H. P.$_d$		Z_S	S	s	Revs.	
								Est.	Act.				Est.	Act.
5	100	.04219	1.41	.2681	.05	.....	1.1	151	.....	1.55	.225	.3422	139	.....
6	140	.05512	1.31	.3217	.067	.....	1.08	186	.....	1.32	.225	.2482	145.9	.....
8	264	.1114	.983	.4289	.089	1.08	1.032	473	464	.975	.225	.2852	204.6	203.4
10	474	.20	.7279	.5361	.145	.87	1.00	914	896	.688	.225	.2846	255.6	253
12	774	.3265	.51	.6433	.255	.62	1.10	1543	1628?	.46	.225	.2842	306.5	303.5

Neither of vessels A and B was employed in obtaining the curves of design but, unfortunately, due to lack of reliable data, they both formed one of several vessels used in obtaining the values of K for their respective types.

Still another vessel, but this time of type 3, will be given as an illustration for that type.

Vessel E.

Hull.

L. L. W. L. = 168′
B = 36′
H = 11.46′
$B \div$ L. L. W. L. = .214
$H \div B$ = .318
L. A. B. = ½ L. L. W. L.
2 L. A. B. $\div H$ = 14.66
Nom. B. C. = .482
K. B. C. = .65
B. C. on $A - B$ = .517
B. C. on $A - B$ cor. for $\frac{H}{B}$ = .517
S. B. C. on W = .646
S = .135

Propeller.

Propellers 1
No. of blades 4
Form ..... Standard
D ..... 9.5′
P ..... 11′
Total $\frac{\text{P. A.}}{\text{D. A.}}$ ..... .338
P. A. $\div$ D. A. ..... .2535
T. S. 6,010
$1 - S$ ..... .865
V 18.91
I. T.$_D$ 2.8
I. H. P. 2,218
P. C. ..... .659
E. H. P. 14.62

Analysis.

v = 13.17 e. h. p. = 696 Z_p = .35 I.H.P.$_d$ = 1228
$\frac{v}{V}$ = .6965 $\frac{\text{e. h. p.}}{\text{E.H.P.}}$ = .4763 K = 1.25 Actual I.H.P.$_d$ = 1208
Z_S = .365
s = .1727
R_d = 146.5
Act. R_d = 146.7

This vessel, also, was not used in obtaining any of the design curves.

To illustrate the method to be followed in making an estimate of the effective horse-power delivered by a propeller when the I. H. P.$_d$ or S. H. P.$_d$ and the speed of the vessel are known, take up the case of vessel F.

Vessel F.—Hull of the dreadnaught type having a standard basic slip of approximately 13 per cent.

Number..4
Blades....3
Form.....Standard
D.....13.42′
P.....15.17′
$\frac{\text{P.A.}}{\text{D.A.}}$..... .3567
T.S.....7760

Propeller.

$1 - S$..... .86
V.....23.97
I.T.$_D$.....4.97
I.H.P....34,259 on four propellers.
P.C..... .648
E.H.P....22,200 on four propellers.
S.H.P....31,519 on four propellers.

Ship steaming with one idle propeller on each side:—

v	7	10	13.5	15
$\frac{v}{V}$	.292	.4172	.5632	.6258
S. H. P.$_d$	1,760	4,443	9,263	11,586
Z_s (curve 1, Fig. 6)	1.44	1.0	.62	.49
s	.3995	.3661	.3182	.295
R_d	77.87	105.4	132.3	142.1
Act. R_d	75.9	101.9	132.1	143.85
S. H. P.	15,760	15,760	15,760	15,760
K	1	1	1	1
$\frac{\text{e. h. p.}_1}{\text{E. H. P.}}$ for $\frac{v}{V}$	.06	.122	.33	.545
Z_1	1.25	.94	.5	.27
Z_p	1.101	.745	.365	.202
e. h. p. ÷ E. H. P.	.105	.192	.442	.635
e. h. p.	1,166	2,131	4,910	7,050

In the above calculations—

$$2Z_p = \text{Log S. H. P.} - \text{Log S. H. P.}_d + Z_1.$$

It has been stated that with vessels of types 1 and 3 the transposition of Z_S from curve 1 to curve 2, Fig. 6, does not occur. This statement requires modification because it does happen when what may be called "induced cavitation" occurs. By this term is meant cavitation which is produced by means other than the conditions normally governing the performance of the propeller. It is frequently encountered with vessels of the destroyer type when propeller struts have been so constructed and fitted as to increase the squat of the hull and to produce areas of reduced pressure in the wake of the struts, the propellers working in these areas. In such cases the value of Z_s gradually passes over from curve 1 to curve 2, the departure from curve 1 for vessels of 310 feet L. W. L. length occurring approximately at a speed $v = 1.42 \sqrt{\text{L. L. W. L.}}$ and arriving on curve 2 when $v = 1.92 \sqrt{\text{L. L. W. L.}}$

To illustrate this phenomenon, the case of a destroyer is given in which the shafts were carried by two struts each, the axis of the lower arm of each forward strut being parallel to the base line of the vessel, while the axes of the lower arms of the after struts were inclined downward at the forward edge, 4½ degrees.

Vessel G.

Hull.	*Propeller.*	
L. L. W. L. = 300.5	Number	2
B = 30.5′	Blades	3
H = 9.583′	Form	Standard
$\frac{B}{\text{L. L. W. L.}}$ = .1015	D	92.5″
Nom. B. C. = .428	P	82″
$\frac{H}{B}$ = .314	$\frac{\text{P. A.}}{\text{D. A.}}$	.595
L. A. B. = 150.25	T. S	13,330
$\frac{2\text{L. A. B.}}{H}$ = 31.35	$P \times R$	3762
B. C. on $C-D$ = .345	$1-S$	.79
B. C. on $C-D$ cor. for $\frac{H}{B}$ = .271	V	29.33
S. B. C. on X = .315	I. $T._D$	11.77
S = .21	I. H. P	18,021
$v_1 = 1.42\sqrt{300.5} = 24.75$	P. C	.525
$v_2 = 1.92\sqrt{300.5} = 33.44$	E. H. P	9,461
$\frac{v_1}{V}$ = .85	S. H. P	15,680
$\frac{v_2}{V}$ = 1.279		

$\frac{\text{e. h. p.}}{\text{E. H. P.}}$	e. h. p.	v	$\frac{v}{V}$	Z_p	K	S. H. P.$_d$		$^*Z_s$	S	s	Revs.	
						Est.	Act.				Est.	Act.
.025	237	10.3	.3512	−1.668	1	356	400	+1.215	.21	.074	165	160
.05	474	12	.4092	−1.355	1	732	720	+1.025	.21	.09823	197.3	200
.075	711	14.3	.4877	−1.1715	1	1117	1150	+ .805	.21	.09093	233.3	230
.10	946	15.52	.5292	−1.0414	1	1507	1500	+ .70	.21	.09568	254.5	252
.20	1892	19.25	.6564	− .7279	1	3102	3100	+ .435	.21	.1070	319.7	317
.30	2838	21.5	.7332	− .5445	1	4732	4750	+ .30	.21	.1196	362.1	359
.40	3785	22.8	.7775	− .4144	1	6385	6300	+ .248	.21	.1432	394.6	394
.50	4730	24	.8184	− .3135	1	8055	8100	+ .165	.21	.1492	418.3	425
.60	5676	25.1	.8559	− .231	1	9741	9700	+ .122	.21	.1634	444.9	453
.70	6623	26.15	.8917	− .1613	1	11436	11500	+ .105	.21	.1845	475.5	476
.80	7570	27.15	.9258	− .1009	1	13143	13100	+ .095	.21	.2072	507.8	508
.90	8514	28.15	.9599	− .0477	1	14855	14900	+ .08	.21	.2277	540.5	538
1.00	9461	29.2	.9957	0.0	1	16580	16550	+ .061	.21	.2417	571	571
1.05	9935	29.94	1.021	+0.0221	1	17445	17400	+ .05	.21	.2479	590.3	591

$^*Z_s$ taken on straight line passing over from $\frac{v_1}{V}$ on curve 1, Fig. 6, to $\frac{v_2}{V}$ on curve 2; where curve of $\frac{\text{e. h. p.}}{\text{E. H. P.}} - \frac{v}{V}$ crosses ordinate of unity load at lower than $\frac{v}{V} = 1$ (Fig. 7), curve 2 becomes a horizontal line coinciding with $Z_s = 0$, Fig. 6. The change of the Z_s condition does not change so abruptly as is done in making the estimate, for inspection of the actual revolutions indicates that it begins at about 23 knots and changes gradually until at 26 knots it has taken its position on the pass-over line used in the estimate.

Cavitation when $\frac{e.\ h.\ p.}{E.\ H.\ P.}$ *Is Greater than Unity.*

When the load fraction exceeds unity, the laws governing the action of the propeller, both as to power and apparent slip increments, appear to change, but the changes occurring appear to be dependent upon the value of the speed fraction at which the performance plots on Fig. 7.

Should the performance plot on or above the curve E, Fig. 7, and at a load fraction exceeding unity, the values of Z_p follow the curve marked 5 on Fig. 6, while should it plot below the curve E, the value Z_p follows the curve 4, the equation for curve 4 being—

$$Z_p = 1.0414 \text{ Log } \left(\frac{\text{e. h. p.}}{\text{E. H. P.}}\right)$$

while that for curve 5 is—

$$Z_p = 1.51 \text{ Log } \frac{\text{e. h. p.}}{\text{E. H. P.}}$$

Should the performance plot above E, the values of Z_s follow the curve 3, which differs slightly in equation from the cavitation curve 2, Fig. 6, the new equation being—

$$Z_s = 3 \text{ Log } \left(\frac{v}{V}\right).$$

Should it plot below E, the values of Z_s, for values of $\frac{v}{V}$ greater than .9385, become zero, the increase in apparent slips being apparently independent of speed but entirely dependent upon power.

In order to illustrate these phenomena, the performances of four vessels will be analyzed, the first, vessel H, having been used in the preparation of all the design curves; vessel I, a reciprocating engine destroyer plotting above E; vessel J, a destroyer plotting below E; vessel K, a dreadnaught plotting below E.

The trial data of vessels I and J are not of the best as the propellers of I were cavitating heavily, producing excessive vibration of the engines and affecting the clearness of the indicator cards, while with vessel J the starboard propeller and torsion meter readings suffered badly through "induced cavitation" of that propeller, the port propeller running smoothly.

VESSEL H. *Type I.*

Hull.	*Propeller.*	
L. L. W. L. = *	Number of propellers	2
B = *	Number of blades	3
H = *	Form of proj. area	Standard
$B \div$ L. L. W. L. = .167	D	18.25′
Nom. B. C. = .6	P	19.75′
B. C. on $C-D$ = .572	P. A. ÷ D. A	.328
$\frac{H}{B}$ = .317	T. S	7,250
B. C. on $C-D$ corrected for $\frac{H}{B}$ = .492	$P \times R$	2,498
Slip B. C. on X = .525	$1-S$	.865
L. A. B. = *	V	21.32
2 L. A. B. ÷ H = 18.88	I. $T._D$	4.35
S = .135	I. H. P	24,800
K = 1	P. C	.665
	E. H. P	16,492

*The data omitted are confidential and therefore cannot be made public.

$\frac{\text{e. h. p.}}{\text{E. H. P.}}$	e. h. p.	v	$\frac{v}{V}$	Z_p	I. H. P.$_d$		Z_S	S	s	Revs.	
					Est.	Act.				Est.	Act.
.0747	1232	9.5	.3896	−1.1718	1670	1600	+1.09	.135	.1118	49.1	52
.09957	1642	10.45	.4902	−1.042	2251	2200	+ .8	.135	.07742	58.12	57.5
.1991	3284	13.21	.6196	− .728	4639	4800	+ .502	.135	.08023	73.69	73.6
.2987	4926	15.05	.7059	− .545	7054	7125	+ .346	.135	.08538	84.43	84.5
.3983	6568	16.45	.7716	− .415	9538	9300	+ .235	.135	.08919	92.66	92.5
.4978	8210	17.7	.8302	− .314	12035	11600	+ .145	.135	.09148	99.96	99.75
.5974	9852	18.92	.8875	− .233	14503	14625	+ .07	.135	.09275	107	106.8
.697	11494	19.76	.9269	− .1615	17098	17200	+ .017	.135	.09679	112.3	112.1
.7965	13136	20.33	.9536	− .1012	19645	19600	+0	.135	.1069	116.8	116.2
.8961	14778	20.8	.9756	− .048	22205	22200	+0	.135	.1209	121.4	120.2
.9957	16420	21.24	.9963	− .001	24740	25200	+0	.135	.1347	125.9	124.8
1.045	17230	21.44	1.006	+ .0294*	26537	26600	+ .015	.135	.1396	127.9	126.5
1.095	18062	21.62	1.014	+ .056*	28213	28100	+ .023	.135	.1457	129.9	129.2

The next vessel given is extremely interesting on account of the enormous range of load fraction and large range of speed fraction through which the performance extends.

VESSEL I. *Type 2.*

Hull.	*Propeller.*	
L. L. W. L. = 245′	Number	2
B = 23.52′	Blades	3
H = 6.687′	Form	Standard
$B \div$ L. L. W. L. = .096	D	7.42′
Nom. B. C. = .41	P	10.833′
B. C. on $C-D$ = .332	P. A. ÷ D. A	.275
$H \div B$ = .284	T. S	6400
B. C. on $C-D$ corrected for $\frac{H}{B}$ = .247	$P \times R$	2974
S. B. C. on X = .295	$1-S$	.7775
2L. A. B. = 245	V	22.82
2L. A. B. ÷ H = 36.64	I. T.$_D$	3.23
S = .2225	I. H. P	3625
	P. C	.693
	E. H. P	2512

*In this case Z_p follows curve 5, Fig. 6, and Z_S follows curve 3.

v	$\frac{v}{V}$	e. h. p.	$\frac{\text{e. h. p.}}{\text{E. H. P.}}$	Z_p	I. H. P.$_d$		Z_S	S	s	Revs.	
					Est.	Act.				Est.	Act.
16	.7013	467	.1859	−.762	627	.....	+.35	.2225	.08617	163.8	.....
18	.789	787	.3133	−.522	1090	.....	+.21	.2225	.1085	188.9	184
20	.8767	1287	.5123	−.298	1825	.....	+.082	.2225	.1353	216.4	220
22	.9643	1978	.7874	−.105	2847	2750	+.0	.2225	.1747	249.4	255
24	1.052	2881	1.147	+.09	4460	4500	−.07	.2225	.233	292.6	296.5
25	1.096	3297	1.313	+.18	5487	5500	−.122	.2225	.2543	313.6	313.5
26	1.14	3738	1.488	+.26	6597	6500	−.178	.2225	.2687	332.6	324
27	1.184	4193	1.669	+.34	7931	7600	−.226	.2225	.2893	355.4	342
28	1.227	4474	1.781	+.38	8696	8750	−.278	.2225	.2814	364.5	368
28.5	1.249	4622	1.84	+.4	9106	9300	−.3	.2225	.2801	370.3	374

The performance of this vessel's propellers enters the overload zone to a greater extent than any other of which the records are available. The maximum load fraction attained is 1.84, while the corresponding speed fraction is 1.249. The ratio S. H. P. = .92 I. H. P. probably does not exactly fit this case as the main air pumps were driven by the main propelling engines and absorbed some of the power developed by those engines.

Vessel J.

This vessel is a destroyer fitted with turbines connected directly to the propeller shafts. The revolutions are high with resultant high tip speeds, while the effective thrusts delivered at 35 knots were also very high. The blades of the starboard propeller had been injured by striking something and were run in this injured condition, with the result that the performance of this propeller was extremely erratic. The port propeller, however, ran smoothly to the end of the trials, and the power data taken from this shaft were consistent. In comparing the estimated and actual performances, the actual power of the port shaft has been doubled for the total power, while the revolutions of that shaft have been taken as the normal revolutions.

Hull.	*Propeller.*	
L. L. W. L. = 310′	Number	2
B = 30.7′	Blades	3
H = 9.146′	Form	Standard
$B \div$ L. L. W. L. = .99	D	7.8′
Nom. B. C. = .476	P	7.42′
$H \div B$ = .298	P. A. ÷ D. A	.648
S. B. C. on X (corrected) = .285	T. S	15,100

2L. A. B.$\div H$=33.89	$P\times R$ 4,572
S=.222	$1-S$ ..... .778
	V 35.11
	I. T.$_D$ 13.65
	I. H. P 26,025
	P. C ..... .525
	E. H. P 13,663
	S. H. P 23,943

Analysis.

v	e. h. p.	$\frac{v}{V}$	$\frac{\text{e. h. p.}}{\text{E. H. P.}}$	Z_p	K	S. H. P.$_d$		Z_S	S	s	Revs.	
						Est.	Act.				Est.	Act.
12	430	.3418	.03147	−1.53	1	707	800	+1.255	.222	.1179	185.7	184
16	1050	.4558	.07685	−1.14	1	1735	1800	+ .894	.222	.1260	249.9	249.9
20	2210	.5697	.1618	− .82	1	3624	3900	+ .62	.222	.1401	317.4	310
24	4700	.6837	.3440	− .481	1	7910	8300	+ .388	.222	.1792	399.1	389
28	8600	.7976	.6294	− .205	1	14934	15000	+ .2	.222	.2195	489.6	468
32	12970	.9115	.9493	− .0221	1	22755	23000	+ .032	.222	.2271	565.1	563
34	15100	.9685	1.105	+ .0632	1	27695	27500	+ .0	.222	.2594	626.6	623
35	16200	.997	1.186	+ .08	1	28953	29400	+ .0	.222	.2669	652	655

These vessels begin squatting at about 20 knots and, when the struts are properly located so that no eddying with consequent reduction of pressure occurs, an increase in wake over the normal is generally encountered, resulting in a slowing down of actual revolutions below the estimate. The tip speed of the propeller at 35 knots was 16,050 feet with an effective thrust per square inch of projected area of 16.9 pounds, both figures being exceptionally high.

Vessel K.

This vessel is of the dreadnaught type and has propellers of such characteristics as to plot her performance considerably below curve *B*, Fig. 7. Her hull is of type 1.

Hull.	*Propeller.*
L. L. W. L.=*	Number 4
B=*	Blades 3
H=*	Form ..... Standard
$B\div$L. L. W. L.=.168	D ..... 9.583′
Nom. B. C.=.622	P ..... 8.193′
B. C. on $C-D$=.586	P. A.$\div$D. A ..... .501

*The data omitted are confidential and therefore cannot be made public.

$H \div B = .305$	T. S. 10,670
B. C. on $C-D$ corrected for $\frac{H}{B} = .455$	$P \times R$ 2,904
S. B. C. on $X = .49$	$1-S$ ..... .84
2L. A. B. $\div H = 19.5$	V 24.13
$S = .16$	I. $T._D$ 8.75
K. B. C. $= .612$	I. H. P 31,732
Mean T. C. Large	P. C ..... .554
$K = 1$	E. H. P 17,580
	S. H. P 29,196

Analysis.

$\frac{\text{e. h. p.}}{\text{E.H.P.}}$	e. h. p.	v	$\frac{v}{V}$	Z_p	K	S. H. $P._d$		Z_S	S	s	Revs.	
						Est.	Act.				Est.	Act.
.1	1758	10.31	.4273	−1.0414	1.0	2654	2750	.98	.16	.139	148	153
.2	3516	12.96	.5371	− .7279	1.0	5463	5400	.68	.16	.143	187	189.5
.3	5274	14.81	.6138	− .5445	1.0	8333	8100	.515	.16	.149	215	218
.4	7032	16.3	.6755	− .4144	1.0	11244	11050	.400	.16	.158	240	241
.5	8790	17.44	.7228	− .3135	1.0	14185	13700	.317	.16	.161	257	259
.6	10548	18.42	.7634	− .231	1.0	17152	16300	.250	.16	.167	274	274.5
.7	12306	19.32	.8007	− .1613	1.0	20138	19150	.19	.16	.171	288	288.5
.8	14064	20.1	.8350	− .1009	1.0	23143	22000	.142	.16	.176	302	302.5
.9	15822	20.7	.8579	− .0477	1.0	26159	25300	.11	.16	.185	314	315
1.0	17580	21.13	.8757	− .00	1.0	29196	28300	.082	.16	.193	324	326
1.05	18459	21.3	.8848	+ .0221	1.0	30720	30000	.075	.16	.200	329	331.5

Concerning the efficiency of propulsion, results of trials indicate that with type 1 and probably very fine type 3 ships, where the basic slip is greater than the S values corresponding to curve B, Fig. 7, there is a gradual increase in efficiency in passing from the upper curve of S to B and then a gradual decrease until the cavitating curve for the basic S is reached. However, in making the estimates of performance, this probable increase should be omitted and carried as "so much possible velvet." The maximum gain is probably reached with basic slip between 21 and 22 per cent, becoming practically zero with those basic slips for which the upper S curves on Fig. 7 approximate closely to B or are below B.

WAKE GAIN AND LOSS.

Before taking up the subject of design, there is one phenomenon met with to which attention should be called, and this is the effect produced by a change of

wake which may occur due to the vessel altering her lines of flotation. This variation of wake may be due to a change in trim, causing an increase in wake during a portion of her career, but the wake usually returning to normal after an upper limiting speed is attained, or it may be caused by the vessel lifting bodily in the water, decreasing its submerged fullness and tending to plane. This latter action is accompanied by a heavy decrease in wake and a corresponding falling off in efficiency in propulsion.

Vessels which slightly alter their trim at a speed depending upon the ratio of v to $\sqrt{\text{L. L. W. L.}}$ may accomplish this change in trim without an increase in the wake. Such a vessel is vessel H, while another ship of similar type may show a decided gain in efficiency during this period of increased wake due to the resulting increased thrust of the propeller produced by the increase in wake. Such a gain is spoken of as "wake gain." Such gain is shown by vessel L, following. Vessels H and L have effective horse-power curves practically identical in form. Both show a slight hump between 15 and 17 knots. The performances on trial of the two ships were radically different. The writer was aboard both of these vessels and noted that vessel H, which was the lighter and smaller of the two ships, after passing 15 knots developed a very peculiar vertical vibration which gave one the impression that she was trying to break her back. This vibration disappeared after passing 17 knots. Vessel L, on the contrary, passed through the range of the hump smoothly and with a total lack of this peculiarity of vibration. Analysis of the performance of vessel L shows that when the estimate of power and revolutions are made for the speeds and effective horse-powers obtained from the model tank, the revolutions obtained agree with the revolutions corresponding to the actual speeds corresponding to these estimated powers.

When a vessel lifts bodily in the water and begins planing as she lifts, the model tank curves indicate a rapid decrease in the increment of increase of the resistance. Should the estimate of power required be made for the e. h. p. and speed given on this curve of deflection, a bitter disappointment will be encountered. Rather, the curve before inflection should be extended by means of a batten and the v's corresponding to this extended curve of e. h. p. will be the actual speeds obtained by using the model tank curve of e. h. p. and v. After planing is accomplished, the wake gradually increases to normal as the speed is increased.

Gain by possible wake increase should never be counted on as a certainty, but loss by wake decrease should always be allowed for.

Vessel L.

In the case of this vessel, the estimated powers and revolutions are those obtained from the model tank curve while there are also given columns of the actual S. H. $P._d$, speeds with corresponding revolutions, obtained by fairing the hump out of the model tank curve and using the new speeds as those to be expected rather than those used in the estimate.

Condition 1.

Hull.	*Propeller.*	
L. L. W. L. = *	Number	4
B = *	Blades	3
H = *	Form	Standard
$B \div$ L. L. W. L. = .162	D	12.863′
Nom. B. C. = .646	P	11.209′
B. C. on $C-D$ = .595	P. A. ÷ D. A	.4247
$H \div B$ = .297	T. S	9,000
B. C. on $C-D$ corrected for $\frac{H}{B}$ = .455	$P \times R$	2,502
S. B. C. on X = .49	$1-S$	.865
L. A. B. = *	V	21.61
2L. A. B. $\div H$ = 20.82	I. $T._D$	6.62
S = .135	I. H. P	37,404
M. T. C. = Large	P. C	.603
K = 1	E. H. P	22,556
	S. H. P	34,412

Analysis.

$\frac{\text{e. h. p.}}{\text{E. H. P.}}$	e. h. p.	v tank	$\frac{v}{V}$	$\sum Z_p$	K	S.H.P.$_d$	v_1 faired	S.H.P.$_d$ for v_1	Z_S	S	s	R_d	
												Est.	Act. for v_1
.1	2256	11	.5091	−1.0414	1	3128	11.25	3250	+.75	.135	.069	106.8	102
.2	4512	13.86	.6415	− .7279	1	6439	14.0	6550	+.46	.135	.0729	135	129.6
.3	6768	15.75	.729	− .5445	1	9822	16.12	9850	+.305	.135	.0778	154.4	151.3
.4	9024	17.25	.7984	− .4144	1	13253	17.67	13200	+.198	.135	.082	169.9	168
.5	11280	18.4	.8517	− .3135	1	16719	18.8	16750	+.118	.135	.0861	182	181.7
.6	13536	19.3	.8933	− .231	1	20217	19.7	20300	+.061	.135	.0913	192	190.6
.7	15788	20.05	.928	− .1613	1	23736	20.34	23700	+.015	.135	.0964	200.6	198.6
.8	18048	20.7	.9581	− .1009	1	27278	20.85	27250	0	.135	.107	209.6	206
.9	20304	21.15	.9789	− .0477	1	30833	21.25	31100	0	.135	.1131	215.6	213.5
1.0	22556	21.45	.9928	0	1	34412	21.55	Off curve	0	.135	.135	224.2	221

DESIGN OF THE PROPELLER.

In treating this subject an attempt will be made to be as brief as possible, confining the discussion entirely to the precautions that must be taken in deciding upon the initial point of the design and to the formulas used in obtaining the design.

*The data omitted are confidential and therefore cannot be made public.

Determination of the Design Point.

The data supplied the designer in order to determine upon the proper propeller to meet any given set of conditions usually come in one of three forms:—

1. Required speed, effective horse-power for this speed, and revolutions of propeller desired at this speed.

2. Effective horse-power curve for the ship, designed power of the engines and revolutions of the propeller at this power.

3. Designed power of the engines and revolutions of the propeller, desired speed of vessel.

With each of these are supplied the hull characteristics of the vessel, together with distance between center of propeller hub and skin of ship or between center of hub and limiting dimensions of the propeller well.

When data of form 1 are given, the problem consists in determining not only the characteristics of the propeller but also the engine power necessary to drive the vessel at the required speed.

When data of form 2 are given, in addition to the characteristics of the propeller, the maximum speed of vessel which can be obtained with the designed power of the engines is also determined.

Finally, with form 3 data, the propeller designer is in the hands of the naval architect. So far as the designer of the propeller is concerned, the ship may or may not make the desired speed with the designed power at the designed condition of load. He can only feel confident that she will make it at some load condition. To call on him for a guaranteed performance is, to say the least, an absurdity, and to blame him for failure is to carry injustice to the extreme.

In selecting the point upon which to base the design, Fig. 7 is used, and the preliminary point or points taken will depend upon the type of vessel and the derived basic slip of the vessel. The point or points taken should be at sufficient height above the cavitation curve for the basic slip and the value of K as to insure that for any probable increase in resistance of the hull, due to deep loading, foul bottom, or ordinary heavy weather conditions, the value of $v \div V$ will not be reduced to such an extent as to lower the performance point below the curve of cavitation.

Ratio of Pitch to Diameter.

The limitation for this ratio is not definitely known, but for safety it is recommended that a lower value of about .75 be taken as a limit, although conditions may arise where it will be found necessary to go even lower than this value. The curve E, Fig. 7, need not be adhered to rigidly as the upper limit of design, but this limit and the maximum limit of $\frac{\text{e. h. p.}}{\text{E. H. P.}}$ used for the adopted point of design should be regulated by the values of $P \div D$ and of P. A. $\div$ D. A. which are obtained in the calculation. It is undesirable to use a basic value of P. A. $\div$ D. A. of less than approximately .23 on account of the bluntness of blade sections which result with the smaller values.

Formulas for Design.

Form 1: Data as per (1).

(1) v (required speed)	v	v	v	(Const.)
(2) $n\times$e. h. p. (required eff. horse-power)	$n\times$e. h. p.	$n\times$e. h. p.	$n\times$e.h. p.	(Const.)
(3) n (number of propellers)	n	n	n	(Const.)
(4) e. h. p. (on 1 propeller)	e. h. p.	e. h. p.	e. h. p.	(Const.)
(5) e. h. p. $\div$ E. H. P. (load fractions used)	Val. 1	Val. 2	Val. 3	(Variable)
(6) $v \div V$ (speed frac. for values of (5))	1	2	3	(Variable)
(7) E. H. P. $= (4) \div (5)$	1	2	3	(Variable)
(8) $V = (1) \div (6)$	1	2	3	(Variable)
(9) S (basic slip from hull characteristics)	S	S	S	(Constant)
(10) D (proposed diameter)	D	D	D	(Constant)
(11) E. $T._p \times$ (P. A. $\div$ D. A.)	1	2	3	(Variable)

$$\text{E. } T._p \times (\text{P. A.} \div \text{D. A.}) = \begin{cases} (2.88 \times \text{E. H. P.}) \div (D^2 \times V) \text{ for 3-bladed propellers.} \\ (2.491 \times \text{E. H. P.}) \div (D^2 \times V) \text{ for 4-bladed propellers.} \\ (3.84 \times \text{E. H. P.}) \div (D^2 \times V) \text{ for 2-bladed propellers.} \end{cases}$$

(12) P. A. $\div$ D. A. (for (11) Fig. 8, Plate 79)	1	2	3	(Variable)

(12) = Projected area ratio if the propeller is 3-bladed = Basic.

(13) 4/3 (P. A. $\div$ D. A.) = (Total proj. area ratio for 4 blades)	1	2	3	(Variable)
(14) 2/3 (P. A. $\div$ D. A.) = (Total proj. area ratio for 2 blades)	1	2	3	(Variable)
(15) P. C. (propul. coef. for (12), (13) or (14))	1	2	3	(Variable)
(16) I. H. P. = E. H. P. $\div$ P. C. = Basic I. H. P	1	2	3	(Variable)
(17) S. H. P. = .92 $\times$ I. H. P. = Basic shaft horse-power	1	2	3	(Variable)
(18) Z_p = Power factor for $\frac{\text{e. h. p.}}{\text{E. H. P.}}$ (Fig. 6)	1	2	3	(Variable)
(19) K = Power augment (Fig. 3)	K	K	K	(Constant)
(20) I. H. $P._d$ or S. H. $P._d$ = (I. H. P. $\div$ 10^{Z_P} or S. H. P. $\div$ 10^{Z_P}) $\times K$ = Est. of power	1	2	3	(Variable)
(21) T. S. = Tip speed for (12)	1	2	3	(Variable)
(22) $1-S = 1-(9)$	$(1-S)$	$(1-S)$	$(1-S)$	(Constant)
(23) $P = (\pi D \times V \times 101.33) \div$ (T. S. $\times$ $(1-S)$) = Pitch	1	2	3	(Variable)
(24) Z_S = Speed factor for $\frac{v}{V}$, Fig. 6, Curve 1	1	2	3	(Variable)

(25) $s=(S\times$I. H. P.$_d\times 10^{z_S})\div$I. H. P. $=(S\times$S. H. P.$_d\times 10^{z_S})\div$S. H. P. $=$Ap. slip.....	1	2	3	(Variable)
(26) $R_d=(v\times 101.33)\div(P\times(1-s))=$ Est. revs.....	1	2	3	(Variable)

Should the revolutions obtained be less than those desired and a good ratio of $P\div D$ exist, the revolutions may be increased in two ways, both of which produce a reduction in efficiency.

(*a*) Retain original diameter and increase values of $v\div V$.

(*b*) Reduce the diameter and retain the original values of $v\div V$.

By the (*a*) process, the projected area ratio will be increased and the values of $P\div D$ and of s will be reduced, but the safety distance above cavitation will increase.

By the (*b*) process, the values of the projected area ratios will be increased, $P\div D$ will remain constant and s will be increased, while the safety remains as originally.

(*c*) A third method may also be used; that is, the rise above the original point may be made by keeping the effective thrust constant. This will maintain constant projected area ratios, reduced efficiency, decrease in s and increase in revolutions, but the reduction in efficiency will not be so great as in (*a*).

Should the revolutions obtained be higher than those desired, follow the opposite course to (*a*), (*b*) or (*c*), and the effects will be opposite. Care must be taken to preserve sufficient safety clearance above the cavitation curve, and to do this it may be necessary to hold the $\frac{v}{V}$ constant and use a lower value of $\frac{\text{e. h. p.}}{\text{E.H.P.}}$

Form 2: Data as per (2).

(1) v (covering range of expected speed)	1	2	3	(Variable)
(2) $n\times$e. h. p. (total ef. H. P. for v's)	1	2	3	(Variable)
(3) $n=$number of propellers.....	n	n	n	(Constant)
(4) e. h. p. $=$ef. H. P. on one propeller.	1	2	3	(Variable)

Proceed as in form 1. Should the revolutions for the designed power be greater or less than those desired, proceed as in (*a*), (*b*) or (*c*).

The expected speed will be that corresponding to the designed power and revolutions as determined by the calculations.

Form 3: Data as per (3).

(1) n S. H. P.$_d$ (designed shaft horsepower.....	n S. H. P.$_d$	n S. H. P.$_d$	n S. H.P.$_d$	(Const.)
(2) n I. H. P.$_d$ (designed I. H. P.) $=$ n S. H. P.$_d\div$.92.....	n I. H. P.$_d$	n I. H. P.$_d$	n I. H. P.$_d$	(Const.)
(3) n (number of propellers).....	n	n	n	(Const.)
(4) I. H. P.$_d$ (I. H. P. on one propeller).	I. H. P.$_d$	I. H. P.$_d$	I. H. P.$_d$	(Const.)
(5) v (expected speed).....	v	v	v	(Const.)

(6) e. h. p. ÷ E. H. P. (assumed load fractions)	1	2	3	(Variable)
(7) $v \div V$ = Speed fractions at assumed points on (6)	1	2	3	(Variable)
(8) D (assumed diameter)	D	D	D	(Constant)
(9) S (basic slip for hull conditions)	S	S	S	(Constant)
(10) $V = v \div (v \div V)$ = Basic speed	1	1	1	(Variable)
(11) K = Power augment	K	K	K	(Constant)
(12) Z_p = Power factor for $\frac{\text{e. h. p.}}{\text{E. H. P.}}$ (Fig. 6)	1	2	3	(Variable)
(13) I. H. P. = I. H. $P._d \times 10^{Z_p} \div K$ = Basic power	1	2	3	(Variable)
(14) I. $T._D \div (1 - S)$	1	2	3	(Variable)

$$\text{I. } T._D \div (1-S) = \begin{cases} (2.88 \times \text{I. H. P.}) \div (D^2 \times V) \text{ for 3-bladed propellers.} \\ (2.491 \times \text{I. H. P.}) \div (D^2 \times V) \text{ for 4-bladed propellers.} \\ (3.84 \times \text{I. H. P.}) \div (D^2 \times V) \text{ for 2-bladed propellers.} \end{cases}$$

(15) P. A. ÷ D. A. = Proj. area ratio for (14) Fig. 8	1	2	3	(Variable)

(15) = Projected area ratio of the propeller if 3-bladed = Basic.

(16) 4/3 (P. A. ÷ D. A.) = Total projected area ratio for 4-bladed.

(17) 2/3 (P. A. ÷ D. A.) = Total projected area ratio for 2-bladed.

(18) P. C. = Propulsive coef. for (15), (16), or (17)	1	2	3	(Variable)
(19) E. H. P. = I. H. P. × P. C. = Basic effective horse-power	1	2	3	(Variable)
(20) e. h. p. = E. H. P. × (e. h. p. ÷ E. H. P.) = ef. horse-power delivered	1	2	3	(Variable)
(21) p. c. = Propulsive coef. expected = $n \times (20) \div$ (1) or (2)	1	2	3	(Variable)
(22) T. S. = Tip speed for (15), Fig. 4	1	2	3	(Variable)
(23) $1 - S = 1 - (9)$	$(1-S)$	$(1-S)$	$(1-S)$	(Constant)
(24) $P = (\pi D \times V \times 101.33) \div$ (T. S. $\times (1 - S)$)	1	2	3	(Variable)
(25) Z_S = Speed factor for $\frac{v}{V}$, Fig. 6, curve 1	1	2	3	(Variable)
(26) $s = (S \times$ (I. H. $P._d$ or S. H. $P._d) \times 10^{Z_S) \div}$ (I. H. P. or S. H. P.)	1	2	3	(Variable)
(27) $R_d = (v \times 101.33) \div (P \times (1 - s))$	1	2	3	(Variable)

If proper revolutions are not obtained, proceed as before.

Attention should be called to one danger that is incurred where I. H. $P._d$ or S. H. $P._d$ are used in obtaining the characteristics of a propeller. This danger is produced by an incorrect estimate of K. Should the K used be too small, the propeller obtained will be too large while the efficiency promised will be greater than will be realized. The propeller will, however, be lifted higher in the safety zone than expected. Should K be too large the opposite results occur.

Propellers of Reduced Diameter.

In many problems that arise, the permissible diameter of propeller will be too small to reduce the revolutions under standard conditions of design to so low a number as the problem requires. In such cases it becomes necessary to resort to what may be called a method of "reduced diameter."

There are two of these methods, as follows:—

1. Method by changing form of blades to "fantail."
2. Method by changing form of blades to "broad tip."

By the first method, the propeller, by using the actual effective horse-power to be delivered or a modification of the propelling I. H. $P._d$ or S. H. $P._d$, is designed for a larger diameter and lower number of revolutions than specified. The tips are then cut off the blades and the original projected areas retained by widening the blade tips beyond the centers of pressure by an amount of area equal to that cut off from the tips. This method produces higher pitches, considerably lower revolutions and slightly less efficiency than the standard method of design and must be used when the reduction in revolutions is considerable.

By the second method, the propeller is designed for a greater diameter than can be carried, using a higher effective or engine power than required for the ship. The tips are then cut off the blades, reducing them to the diameter allowed.

This method maintains practically constant efficiency with the basic full diameter propeller, with a higher pitch and lower revolutions than would be obtained by the regular method of design, but the revolutions are not decreased to the same extent as by the first method of reduction.

Formulas for Reduced Diameter.

Form 1: "Fantail."

(1) D = Diam. possible	D	D	D (Constant)
(2) D_1 = Assumed diameter > D	1	2	3 (Variable)
(3) $D_1 \div D$	1	2	3 (Variable)
(4) $(D_1 \div D)^3$	1	2	3 (Variable)
(5) e. h. p. (effective horse-power on one propeller)	e. h. p.	e. h. p.	e. h. p. (Constant)

Now proceed as in preceding methods, using D_1, until I. H. $P._d{}^1$ or S. H. $P._d{}^1$ have been obtained. The actual corrected power will then be—

$$\text{I. H. } P._d \text{ (corrected)} = \text{I. H. } P._d{}^1.$$

Use D_1 in obtaining the pitch, and then proceeding to the apparent slip, use the uncorrected value of I. H. $P._d{}^1$ or S. H. $.P_d{}^1$. Use this slip in obtaining the revolutions for the propeller with diameter D_1, then corrected revolutions equal—

$$R_d \text{ (corrected)} = R_d{}^1 \times \left(\frac{D_1}{D}\right).$$

When the basis of design is engine power instead of effective, the power must be reduced as follows:

$$\text{I. H. } P._d{}^1 = \text{engine power.}$$

Form 2: "Broad Tip."

(1) D = Diameter possible.....	D	D	D (Constant)
(2) D_1 = Assumed diameter > D.....	1	2	3 (Variable)
(3) $(D_1 \div D)^2$	1	2	3 (Variable)
(4) $(D_1 \div D)^{\frac{1}{2}}$.....	1	2	3 (Variable)
(5) e. h. p_1. = e. h. p. (actual $\times (D_l \div D)^2$.	1	2	3 (Variable)

Use D_1 and e. h. $p._1$ in designing the full diameter propeller. Calling the speed of ship v, then by the regular method of design, using e. h. $p._1$ for v, the derived power for the full diameter propeller required for the speed v will be I. H. $P._{d1}$ and the actual power required for the reduced diameter propeller will be—

$$\text{I. H. } P._d = \text{I. H. } P._d{}^1 \times \left(\frac{D_1}{D}\right)^2.$$

The actual revolutions will be—

$$R_d = R_{d1}' \text{ (estimated revs., full diam.)} \times \left(\frac{D_1}{D}\right)^{\frac{1}{2}}.$$

Where the engine power forms the basis of the design, this power should be multiplied by $\left(\frac{D_1}{D}\right)^2$ for the power to use in obtaining the full diameter screw.

Such broad tip propellers have one peculiarity, so far as can be ascertained, and that is encountered when they are used on submarines having slips of the first order:

When running on the surface, the revolution correcter is $\left(\frac{D_1}{D}\right)^{\frac{1}{2}}$ as given above. When running submerged and plotting below curve B, Fig. 7, this factor changes to $\left(\frac{D_1}{D}\right)^{\frac{1}{3}}$.

CONCLUSION.

The paper as submitted embraces the results of many years of study by the author so far as what may be considered orthodox forms of hulls with orthodox propeller conditions are concerned. The questions of small diameter propellers

carried as deeply as possible in the cases of deep draught vessels, propellers for double enders, propellers for tunnel boats, etc., have not been touched. All of these various problems require special study to determine the particular values of K and of S to use in the designs of the propellers.

The problems involved in so recording the results and conclusions of years of study and experience in such a manner that all may readily make use of them have not been easy ones, and even in the final results as submitted there are ranges of indefiniteness which the author, due to lack of reliable data, has no means of clearing up.

Several of these regions of indefiniteness are the determination of the ranges of fineness of hulls covering the transition from hulls of type 2 to those of type 1, and from those of type 1 to those of type 3, also the slight range of speed fraction values at the curve B, Fig. 7, in which the apparent slips pass from slips of the first order to those of the second.

Commander McEntee, with his hull models driven by model propellers, has provided a means by which some of the problems may be solved and by which, when propellers are designed in the safety zones, the true values of K may be obtained, but it is stated that this method of Commander McEntee's is of no use in the cavitation ranges, although the author does not agree with this statement where "Dispersal of the Thrust Column" occurs.

For a complete solution and accurate representation of the phenomena that occur, it is necessary to look to the future, trusting that it will bring with it a vast amount of performance data of sufficient value that it can be used with confidence, and the complete life history of the hull with its propeller plotted in such a manner as to provide an infallible guide to success in the solution of later problems.

It is hoped that the material which has been opened up for inspection will thoroughly impress the student with the fact that the hull with its propeller must be treated in conjunction in the solution of the problem of propulsion. To treat of the propeller without its accompanying hull is to lose time in the ultimate complete solution of the problem involved. The perfect solution of the "Problem of the Hull and its Screw Propeller" rests in the hands of the shipowners, naval architects and marine engineers, and without their hearty and intelligent cooperation in the future, the problem may never be entirely solved.

DISCUSSION.

THE CHAIRMAN:—Paper No. 10, "The Problem of the Hull and Its Screw Propeller," by Rear Admiral Dyson, is open for discussion. Will Commander McEntee discuss this paper?

COMMANDER WILLIAM MCENTEE, *Member*:—There are two general methods of

dealing with the propeller: (1) The theoretical method, which is based entirely on equations or formulae deduced by applying accepted laws of physics and using the fundamental units of length, mass and time. (2) The empirical method, in which the relation between the many variables upon which the functioning of the propeller depends is derived from actual experience with real propellers on real ships.

Admiral Dyson has used the second method and obtains results which are entirely satisfactory so long as used within the range of variables or data on which the method is based, though it requires skill and experience to pick out the proper coefficients to use in a particular case. That the method is not of universal application is readily shown by applying it to a ship and its model. I have done this in the case of the vessel L, the data for which are given on page 206 of the paper. The results for the ship agree very well with the design curves given in Plate 75. Thus on the ship for projected area ratio of .4247 at 223 revolutions per minute the tip speed is 9,000 feet per minute, the indicated thrust is 6.62 pounds per square inch of the projected area, and the propulsive coefficient is .603. I applied the same curves to the model of this ship which we tested in the model basin. Of course the model tip speed will be much less, also the pressure per square inch in the projected area is much less, so that if these curves are truly universal, they should fit the model as well as the ship. They go away off the curves, bringing the projected area ratio below anything which was possible to use in the propeller, that is to say, as far down as .05, which would make a very narrow and blunt blade.

On the model of this ship at corresponding speeds with the same projected area ratio the model propellers make 1,230 revolutions per minute with a tip speed of 1,640 feet per minute and an indicated thrust of 1.2 pounds with a propulsive coefficient slightly less than for the ship.

By referring to Plate 75, it will be seen that this tip speed corresponds to a P.A. ÷ D.A. = .025 and P.C. = .525, while indicated thrust per square inch of projected area corresponds to a P.A. ÷ D.A. = .157 — and P.C. = .71.

As the model, propeller and appendages are exactly similar in form to the ship, the factors which depend upon the interaction between the ship and propeller must be the same for model and ship. I think it is a fair conclusion that the fundamental design data given in Plate 75 which is found to agree with the ship data and to disagree decidedly with the model data must therefore be considered to be applicable to a limited field only. It is practically universally admitted now among naval architects that the laws connecting the resistance of a ship and its model are known with sufficient accuracy to permit the prediction of the effective horse-power of the ship from model tests. It is only one step further to find the laws connecting the shaft horse-power and the revolutions per minute for the model and the shaft horse-power and revolutions per minute for the ship. This step has, I believe, already been taken. In tests made under my direction at the Washington Model Basin on a 20-foot model of the battleship New Mexico, it was found that when the revolutions per minute and shaft horse-power as obtained by tests of the model were extended to the ship they agreed with the ship trial data within less than 1 per cent over a range of ship speeds from 10 to 21 knots.

The extensions from the model to the ship were made by using the principle of mechanical similitude or law of comparison and by assuming the thrust deduction and wake coefficients to be the same for both ship and model.

When it is considered that the ship is thirty times as long as the model and that 30,000 shaft horse-power on the ship was derived from about two-tenths of a shaft horse-

power in the model, the close agreement between the performance of the ship on trial and the estimates from the model tests must be taken as a remarkable confirmation of the conclusion that the law of comparison applies to propellers as actually fitted on ships.

Now it must be admitted that the foregoing results have been obtained by the application of theory and, as has been shown before, the extensions from the model to the ship could not have been easily made by empirical methods and certainly not by the particular method used by Admiral Dyson.

I do not mean to say by this that Admiral Dyson's method has not given valuable results in designing propellers for ships, but I think that the enormous amount of valuable data which he has accumulated and which is presented in his paper could be made more readily available to the ordinary man if it were coupled with a theory such as that of mechanical similitude.

It seems to me that the next step in the development of our knowledge of the action of propellers, and the putting of this knowledge into suitable form, would be to find the factors connecting the formulae derived by theoretical means with the empirical facts of experience. When this is done I believe that the design of propellers will be very much simplified, and the student will have fewer obstacles to contend with in acquiring a sufficient knowledge of the subject. What is really needed is not a system of propeller design based on theory alone, nor one which is based on empirical methods alone, but one which will combine the good features of each method as well as eliminate the weakness of each.

I trust the author of this paper, who has deservedly obtained a wide reputation and is rightfully considered to be one of our leading authorities on the subject, may find time to connect his empirical method with sufficient theory to accomplish a happy blending of theory and practice and thus make his results more readily accessible to naval architects and marine engineers.

Mr. E. A. Stevens, Jr., *Member*:—For the past eight years I have followed with no small amount of interest various articles published in the Journal of the American Society of Naval Engineers, written by Admiral Dyson, and have noticed the advances made from time to time, especially regarding the determination of the slip block coefficient and the Power Augment Factor K. I must confess that I was somewhat surprised to see the difference in the way of determining these factors (especially the S. B. C.) in this article and the method as suggested in the Marine Engineers' Handbook, in which no attempt is made to correct for the ratio of draught to beam ($H \div B$).

I have not had time to study this paper carefully but, on looking over it, the following points appear to need a little explanation:—

1. Is it to be understood that, in determining the slip B. C. of single-screw ships of all three types, no correction for variation of midship-section coefficient is to be made? And does this also apply to twin-screw ships of types 1 and 2?

2. Is there no correction to be made for variation of midship-section coefficient in arriving at the K block coefficient?

3. Would it not be as accurate and less complicated to use the prismatic in place of the block coefficient?

4. To what type would high-speed yachts, torpedo boats, and the earlier destroyers belong? The fining of the after body of these vessels is produced neither by a rapid dead rise of the after sections nor by a rapid decrease of beam, but by the raising of the

keel. As the propellers of these vessels are, to a great extent, covered by the submerged hull, the inference might be that they belong to type 1.

5. What is meant by a full midship section? Is it one that is as full or fuller than the standard as shown on Plate 71, and is a fine midship section one that is considerable finer than this standard?

6. How would Plate 76 be applied to ships of type 2? Although one of this type is shown in Problem 1, the method of getting the slip is not explained.

For a comparison between the results obtained by the methods as shown in this paper and in Marine Engineers' Handbook, I have taken the third ship in Problem 1. The calculations are shown below. The first column is taken from this paper; the second is as calculated by the latter method.

S. B. C.	=	.825	.516	P. C.	=	.696	.696
K. B. C.	=	.786	.76	E. H. P.	=	3575	3485
Blades	=	3	3	S. H. P.	=	4726	4600
D	=	16.583	16.583	v	=	11	11
P	=	14.75	14.75	$v \div V$	=	.7614	.755
$\frac{\text{P. A.}}{\text{D.A.}}$	=	.267	.267	e.h.p.	=	1260	1260
T.S.	=	6230	5900	e.h.p. $\div$ E.H.P.	=	.3525	.361
$P \times R$	=	1764	1670	Z_p	=	$-.47$	2.89*
$I-S$	=	.83	.885	K	=	1.23	1.18
V	=	14.45	14.58	S.H.P.$_d$	=	1970	1880
I.T.$_d$	=	3.09	3.07	$\frac{v}{V}$	=	.7614	.755
I.H.P.	=	5137	5010	Z_s	= plus	.25	2.89*
				s	=	.126	.109
				R	=	86.46	84.8

*$2.89 = 10^z$

It would appear that when a propeller is to be designed for a ship, where the effective horse-power is not given but has to be estimated, or if the value of $2\text{L.A.B.} \div H$ is not known, the original method is about as good, at least for ships of this class. For small single-screw steam yachts, with a large ratio of beam to length and with small midship-section coefficient, the method as shown in this paper would, I believe, be decidedly more accurate.

Before closing I wish to state that the foregoing questions are asked in order to obtain information on the above points, as there is some doubt in my mind, also that Admiral Dyson deserves the thanks of the profession, not only for this very valuable paper but also for various previous articles that he has written.

Mr. W. W. Smith, *Member*:—Admiral Dyson's paper is of very great interest, and his work has been of great value to shipbuilders. At the Federal Shipbuilding Company we have been using Admiral Dyson's methods of propeller design and have found certain difficulties which I desire to point out.

First, the method is rather difficult to comprehend and use by the average engineer.

Second, it is somewhat difficult to follow cause and effect in the design.

The present method appears to be somewhat simpler than the one we have been using. Mr. McEntee has already called attention to some of the points which I have noted, but I will give the shipbuilders' point of view in regard to these. It is appreciated that propeller designs and investigations are intricate problems involving many variables; but, if it were possible, it would be advantageous, first, to simplify the method, and second, to connect it more directly with basic science, so that cause and effect in the design can be followed and kept track of better by the designer.

Admiral Taylor's method is based directly on the scientific investigation, and the cause and effect can easily be followed. However, Admiral Taylor's curves and data are not connected with actual performance closely enough, and their value is greatly reduced by not having a connecting link between theory and practice. For Admiral Taylor's method more accurate data as to wake and thrust deduction factors would be valuable.

The great advantage of Admiral Dyson's method is that it is based on actual results of the performance of vessels, and therefore gives thoroughly practical results. It is needless to say that any method which does not give such results cannot be used with assurance. It would appear that both Admiral Taylor's and Admiral Dyson's methods have distinct advantages. If the advantages of both could be combined, I think it would give an ideal basis for designing and investigating propellers.

THE CHAIRMAN:—Is there any further discussion? Has Mr. Wetherbee any remarks he desires to make on this subject?

MR. CHARLES P. WETHERBEE, *Member of Council*:—I have nothing to say except to add my testimony to those who have spoken of our great appreciation of what Admiral Dyson has done in the investigation of propellers. I do not think I can make any comments that would be worth taking the time of the Society.

THE CHAIRMAN:—I may say, in view of the many inquiries which have been made of Admiral Dyson, that he may find it more desirable to make some of his replies in the form of a written discussion after he has had opportunity to see exactly what the comments that have been offered mean. It is difficult to follow them in a verbal discussion like this.

REAR ADMIRAL DYSON:—It appears from Commander McEntee's remarks that his only criticism of the results of my years of labor is that they are limited in their range, that while they will produce correct results for the actual ship, a propeller corresponding to the model propellers used by him in the model tank cannot be produced by the use of the charts of design. He states that he has attempted to produce such propellers and the resultant propellers have had blades of extremely narrow form; in fact, so narrow as not to be practical.

The answer to this criticism of Commander McEntee's is a very simple one. Commander McEntee does not know how to handle the charts to obtain the model propeller. He evidently entered them with the same load fraction, $\frac{\text{e. h. p.}}{\text{E.H.P.}}$, and the same speed fraction, $\frac{v}{V}$, as were used in designing the original propellers for the ship. This would

result, when the projected area ratio of the model screw was kept equal to that of the original propeller, in a propeller of less diameter than his model propeller, of very much lower pitch and of exceedingly high revolutions. Should he hold the diameter equal to that of his model screw, the pitch obtained will be much lower than that of the model and the projected area ratio much smaller.

Should he, however, apply the law of comparison to the factors used in designing the propeller for the ship, assuming that wake remains constant and that the frictional resistance of the hull follows the law of comparison, which it does not, he would obtain the proper factors for use in designing the model propeller, as, for example:—

	Ship	*Ratio*	*Model*
Length on load water line	L. L. W. L.	R	$\frac{\text{L. L. W. L.}}{R}$
Speed of ship	v	$\frac{1}{R^{1/2}}$	$\frac{v}{R^{1/2}}$
Effective horse-power for v	$e.h.p.$	$\frac{1}{R^{7/2}}$	$\frac{\text{e. h. p.}}{R^{7/2}}$
Basic effective horse-power	E. H. P.	$\frac{1}{R^2}$	$\frac{\text{E. H. P.}}{R^2}$
Basic horse-power (engine or shaft)	H. P.	$\frac{1}{R^2}$	$\frac{\text{H. P.}}{R^2}$
Load fraction	$\frac{e.h.p.}{\text{E. H. P.}}$	$\frac{1}{R^{3/2}}$	$\frac{\text{e. h. p.}}{R^{3/2} \times \text{E. H. P.}}$
Speed fraction	$\frac{v}{V}$	$\frac{1}{R^{1/2}}$	$\frac{v}{R^{1/2}\,V}$
Basic apparent slip	S	1	S
Diameter of propeller	D	$\frac{1}{R}$	$\frac{D}{R}$
Pitch of propeller	P	$\frac{1}{R}$	$\frac{P}{R}$
Projected area ratio of propeller	$\frac{\text{P. A.}}{\text{D. A.}}$	1	$\frac{\text{P. A.}}{\text{D. A.}}$

Taking these values in the equations for power; for ship:—

(1) $$\text{Log H. P.}_d = \text{Log I. H. P.} - Z_{pg}$$

For model:

$$\text{Log h. p.}_d = \text{Log h. p.} - Z_{pg}$$

$$Z_{pg} = 1.0414 \text{ Log}\left(\frac{\text{E. H. P.}}{\text{e. h. p.}}\right) \text{ for ship.}$$

$$Z_{pg} = 1.0414 \text{ Log}\left(\frac{\text{E. H. P.}}{\text{e. h. p.}} \times \frac{R^{7/2}}{R^2}\right) = 1.0414 \text{ Log}\left(\frac{\text{E. H. P.}}{\text{e. h. p.}}\right) + 1.0414 \times \frac{3}{2} \text{ Log } R$$

$$(2) \qquad \therefore \text{Log h. p.}_d = \text{Log H. P.} - 2\text{Log } R - 1.0414 \text{ Log}\left(\frac{\text{E. H. P.}}{\text{e. h. p.}}\right) - 1.5621\ R$$

$$(3) \qquad \text{Log H. P.}_d = \text{Log H. P.} - 1.0414 \text{ Log}\left(\frac{\text{E. H. P.}}{\text{e. h. p.}}\right)$$

Subtracting (3) from (2):

$$\text{Log h. p.}_d = \text{Log H. P.}_d - 3.5621R$$

$$\text{Log h. p.}_d = \text{Log H. P.}_d - 3.5621R$$

$$\therefore \text{h. p.}_d = \frac{\text{H.P.}_d}{R^{3.5621}}$$

Turning now to the apparent slips and revolutions,

For ship:

$$s = S\ \frac{\text{H. P.}_d \times 10^{Z_s}}{\text{H. P.}}$$

For model:

$$s = S\ \frac{\text{h. p.}_d \times 10^{Z_s}}{\text{h. p.}}$$

For ship:

$$Z_s = 2.861 \text{ Log}\left(\frac{V}{v}\right) - .0788.$$

For model:

$$Z_s = 2.861 \text{ Log}\left(\frac{V}{v} \times R^{1/2}\right) - .0788.$$

$$\text{h. p.}_d = \frac{\text{H. P.}_d}{R^{3.5621}}$$

$$\text{h. p.} = \frac{\text{H. P.}}{R^2}$$

$$(4) \qquad \text{Log } s = \text{Log } S + \text{Log H.P.}_d + 2.861 \text{ Log}\left(\frac{V}{v}\right) - .0788 - \text{Log H. P.}$$

$$(5) \qquad \text{Log } s = \text{Log } S + \text{Log H.P.}_d - 3.5621 \text{ Log } R + 2.861 \text{ Log}\left(\frac{V}{v}\right) + 2.861 \times 1/2 \text{ Log } R$$
$$- .0788 - \text{Log H. P.} + 2 \text{ Log } R.$$

Subtracting (4) from (5):—

$$\text{Log } s - \text{Log } s = -3.5621 \text{ Log } R + 1.4305 \text{ Log } R + 2 \text{ Log } R.$$

$$\text{Log } s = \text{Log } s - .1316 \text{ Log } R.$$

$$s = \frac{s}{R^{.1316}}$$

For ship:

$$R_d = \frac{v \times 101.33}{P \times (1-s)}$$

For model:

$$r_d = \frac{v \times 101.33}{R^{1/2} \times \frac{P}{R} \times \left(1 - \frac{s}{R^{.1316}}\right)} = \frac{R^{.6316}\,(v \times 101.33)}{P \times (R^{.1316} - s)}$$

from which by assuming any desired value of R and the known value s from trials of of the ship, the revolutions for the model can be obtained, or vice versa.

$$r_d = \frac{R_d \times R^{.6316}\,(1-s)}{R^{.1316} - s}$$

To obtain the apparent slip and revolutions of the actual propeller from the performance of the model, the equations become

$$s = s\,R^{.1316}$$

and

$$R_d = \frac{v \times 101.33}{P\,(1 - s\,R^{.1316})}$$

$$r_d = \frac{v \times 101.33}{R^{1/2} \times \frac{P}{R} \times (1-s)} = \frac{R^{1/2}\,(v \times 101.33)}{P \times (1-s)}$$

$$R_d = r_d \frac{(1-s)}{R^{1/2}\,(1 - s\,R^{.1316})}$$

depending for its value on R and s.

The actual powers and revolutions for the model will probably be higher than those obtained by these equations as the actual effective horse-power is greater than that obtained by the law of comparison and the wake of the model is probably different from that of the ship.

To illustrate the above take first the vessel in the first column of Problem 1 and deduce the model propeller and its performance from the actual by the above formulas, then analyze and estimate the performance of the model screw by the ordinary method.

Conditions of Problem:—Speed of ship = 11 knots; e. h. p. = 1160; R = 30

Analysis	*Comparison*	*Analysis*
$D = 16.583'$	$d = .5528'$	
$P = 14.75'$	$p = .492'$	
$\frac{\text{P. A.}}{\text{D. A.}} = .267$	$\frac{\text{P. A.}}{\text{D. A.}} = .267$	
T. S. = 6230	T. S. = 6230	
$1-S = .86$	$1-S = .86$	
$V = 14.97$	$V = 14.97$	
I. T.$_D$ = 3.09	I. T.$_D$ = 3.09	
I. H. P. = 5137	i. h. p. = $\frac{5137}{R^2} = 5.708$	5.7072
P. C. = 696	P. C. = .696	.696

Analysis	*Comparison*	*Analysis*
E. H. P. = 3575	$E.H.P. = 3.9722$	3.9722
S. H. P. = 4726	s. h. p. = 5.251	5.2506
e. h. p. = 1160	$\text{e. h. p.} = \frac{1160}{R^{7/2}} = .007844$	.007844
$\frac{\text{e. h. p.}}{\text{E. H. P.}} = .3244$	$\frac{\text{e. h. p.}}{E.H.P.} = \frac{.3244}{R^{3/2}} = .001974$	.00197475
$Z_p = 1.0414 \text{ Log } \frac{\text{E. H. P.}}{\text{e. h. p.}}$	$= Z_p = 1.0414 \text{ Log } \left(\frac{E.H.P.}{\text{e. h. p.}} \times R^{3/2}\right) = 2.8164$	
$K = 1.175$	$K = 1.175$	1.175
S. H. P.$_d$ = 1716	s. h. p.$_d = \frac{1716}{R^{3.5621}} = .0093944$	.0094156
$v = 11$	$v = \frac{11}{R^{1/2}} = 2.0083$	2.0083
$\frac{v}{V} = .7348$	$\frac{v}{V} = \frac{.7348}{R^{1/2}} = .13415$	.13417
$Z_s = 2.861 \text{ Log } \left(\frac{v}{V}\right) - .0788 = .30409$	$Z_s = 2.4170$	2.4170
$s = .1024$	$s = .065442$	.06558
$R_d = 84.19$	$R_d = 84.19 \frac{R^{.6316}(1-s)}{R^{.1316} - s} = 442.9$	442.96

Attention is called to the extremely low load and power fractions under which the model screw operates at comparative speeds and loads. Commander McEntee's estimates for the ship are 1750 S.H.P.$_d$ and 84.5 revolutions.

Now take another case, beginning with the model screw, and estimate the power and revolutions for the ship by comparison and by analysis of the actual propeller.

Let us take the case of Vessel H.

Speed of ship	= 21 knots.	Diam. of Propeller	= 18.25′
e.h.p.	= 15575	Pitch of Propeller	= 19.75′
Propellers	= 2 three-bladed.	$\frac{\text{P. A.}}{\text{D. A.}}$ of Propeller	= .328
R	= 30		

Data	Model	Ship by comparison	Ship by analysis
D	.60833′	18.25′	18.25′
P	.65833′	19.75′	19.75′
$\frac{\text{P.A.}}{\text{D.A.}}$	.328	.328	.328
T.S.	7250.		7250.
1-S	.865		.865
V	21.32		21.32
I.T.$_D$	4.35		4.35
I.H.P. (2 Prop.)	27.557		24800.
P.C.	.665		.665
E.H.P.	18.3255		16492.
v	3.8341		21.
$\frac{v}{V}$	.17984		.98504
e.h.p.	.10593		15575.
$\frac{\text{e.h.p.}}{\text{E.H.P.}}$	.0057804		.94441
Z_p	−2.3307		−.025868
K	1.	1.	1.
I.H.P.$_d$	.128685	23506.	23366. (Actual 23400)
Z_s	2.05305		.0
s	.071233		.12719
R_d	635.41	121.26	123.4 (Actual 122.1)

The foregoing problems demonstrate that the model screw performance may be estimated either by analysis of the model screw itself or by the application of the laws of comparison.

Now to design the model propeller by fixing the power and speed fractions to be used in the design, take the propeller of the last problem.

$$D=18.25'; \; P=19.75'; \; \text{P.A.} \div \text{D.A.}=.328; \; v=21; \; \text{e.h.p.}=15575.$$

$$\frac{\text{e. h. p.}}{\text{E.H.P.}}=.96441; \frac{v}{V}=.98504; \; \text{power on two propellers}; \; S=.135$$

For the model, the hull is to be one-thirtieth as long on the L.W.L. as the actual vessel.

$$\therefore R=30; \; d=\frac{18'.25}{30}; \; \frac{v}{V}=\frac{.98504}{\sqrt{30}}=.17984; \; \frac{\text{e. h. p.}}{E.H.P.}=\frac{.94441}{\sqrt{30^3}}=.0057804$$

$$\text{e. h. p.} = \frac{15575}{\sqrt{30^7}} = .10593$$

$$\frac{\text{e. h. p.}}{E.H.P.} = .0057804$$

$E.H.P.$ = 18.3255 (2 propellers)

$$\frac{v}{V} = .17984$$

$$v = \frac{21}{\sqrt{30}} = 3.8341$$

$V = 21.32$

$$D = \frac{18.25}{30} = .60833'$$

$$\text{E. T.}_{\not p} \times \frac{\text{P. A.}}{\text{D.A.}} = 3.3369$$

$$\frac{\text{P. A.}}{\text{D.A.}} = .328 \qquad \text{From Fig. 8.}$$

P. C. = .665

i. h. p. = 27.557

T. S. = 7250

$1-S = .865$

$$P = .65835' = \frac{19.75'}{30}$$

As the model propeller performance can be analyzed by the laws of comparison and as the model propeller can be designed by the use of these laws, it appears that the laws of similitude not only apply to the design charts but also to the propeller itself, and that if such conformity on the part of the charts is to be taken as the proof of their correctness, then judgment in their favor must be given.

When the propeller is working in the range of "cavitation of the thrust column," the estimate of power and revolutions of the actual propeller without cavitation must first be obtained by comparison with the model propeller and the corrections for cavitation applied to these results to obtain the final results; this is necessary as the curves of cavitating thrusts on the chart are not extended down to such low load and speed fractions as those under which the model operates.

Mr. Smith in his discussion expressed the hope that in time the true theory of the propeller may be developed from these charts and a composite method, built up from the work of Admiral Taylor and the author, be produced. The author does not hope for this but is of the opinion that his work combined with that of Commander McEntee forms a complete system of design and checkage as they stand today, the propellers designed by the author's method to be checked in the model tank by the use of the ship's model and the model of the propeller resulting from reduction in size according to the laws of comparison. This combination fills a long-felt want as in the checkage, the model trials give absolute assurance as to the performance of the actual propeller, and enables errors in estimate of K and of S to be corrected, something which has not been had in the past until the actual ship with its propeller has been tried.

In reply to the questions put by Mr. Stevens, the questions will be taken up in order, as follows:

1. Is it to be understood that, in determining the slip B. C. of single-screw ships of all three types, no correction for variation of midship section coefficient is to be made? And does this also apply to twin-screw ships of types 1 and 2?

Answer: No correction is made for type 3 nor for either of types 1 and 2.

2. Is there no correction to be made for variation of midship-section coefficient in arriving at the K block coefficient?

Answer: No correction is made.

3. Would it not be as accurate and less complicated to use the prismatic in place of the block coefficient?

Answer: I do not believe complication would be any the less should the prismatic in place of the block coefficient be used, as in either case we would have to apply the correction for variation of ratios of draught to beam for ratio of length of after body to draught and for length of middle body.

4. To what type would high-speed yachts, torpedo boats, and the earlier destroyers belong? The fining of the after body of these vessels is produced neither by a rapid dead rise of the after sections nor by a rapid decrease of beam, but by the raising of the keel. As the propellers of these vessels are, to a great extent, covered by the submerged hull, the inference might be that they belong to type 1.

Answer: Practically all of these boats are of type 1.

5. What is meant by a full midship section? Is it one that is as full or fuller than the standard as shown on Plate 71; and is a fine midship section one that is considerably finer than this standard?

Answer: A full midship section is one that is as full or fuller than the standard as shown on Plate 71. A fine midship section is one that is considerably finer than this standard. The curve of midship sections on Plate 71 fits vessels of types 1 and 2.

6. How would Plate 76 be applied to ships of type 2? Although one of this type is shown in Problem 1, the method of getting the slip is not explained.

Answer: The curve marked "Slips for S.B.C. on $X-W$, Types 1 and 3 Hulls" has marked on it values of S.B.C. To find the basic apparent slip for any S.B.C. with any given ratio of length of after body to H, pass a direction curve through the point for this S.B.C. as marked on the first-mentioned curve. Where this direction curve intersects the ordinate through the given ratio will be the basic apparent slip to use.

Mr. Stevens says: "It would appear that when a propeller is to be designed for a ship, where the effective horse-power is not given but has to be estimated, or if the value of 2 L.A.B. $\div H$ is not known, the original method (of design) is about as good (as the new), at least for ships of this class. For small single-screw steam yachts, with a large ratio of beam to length and with small midship-section coefficient, the method as shown in this paper would, I believe, be decidedly more accurate."

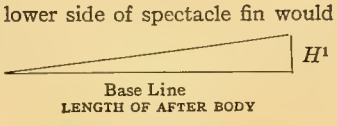
My experience up to date has indicated that the second method gives approximately the same results as to power as were obtained by my original method but it is considered more accurate in determination of revolutions and in designing the screw so as to plot more surely in the safety zone.

Where the value of twice the length of after body divided by H is not known and we have reason to believe that the after body is not abnormally long as compared to the draught nor abnormally short as compared to the draught, the two cross curves of Plate 76 marked for vessels of type 2 and vessels of types 1 and 3 can be used.

Following is given a table of ship characteristics with the type to which each vessel belongs, and as performances of different vessels become available their screws should be analyzed in order to determine accurately the type the hulls belong to and add to the table. By doing so it will be found that it will aid very much in the future, as there are many cases where it is quite difficult to determine whether a vessel is of type 1 or type 3, or whether she is type 2 or type 1, although the latter distinction is easier

Vessel	L.L.W.L. or B.P.	B	H mean	H^1	Nom. B.C.	$H \div B$	2 L.A.B. $\div H$ or H^1	Coef $\int$	Type
	Feet	*Feet*	*Feet*	*Feet*					
A.....	450.	60.083	26.	26.	.6266	.4327	17.31	.956	1
B.....	510.2	85.21	27.02	27.02	.600	.317	18.88	.978	1
C.....	450.	80.22	25.02	25.02	.633	.312	17.99	.943	1
D.....	435.	76.21	23.75	23.75	.656	.311	14.65	.906	3
E.....	528.	62.23	27.5	27.5	.726	.442	13.38	.984	2
F.....	310.	30.7	9.146	9.146	.476	.298	33.89	.756	1
G.....	550.	55.	13.5	13.5	.608	.246	40.74	.942	3
H.....	850.	105.75	31.	31.	.546	.2932	27.42	.957	3
I.....	203.75	19.	16.1	15.1	.481	.847	13.5	.846	2
J.....	268.57	22.16	14.16	11.08	.459	.684	24.24	.751	2–approx 1
K.....	175.	34.125	12.64	12.64	.531	.366	14.81	.875	3
L.....	180.	35.33	13.23	13.23	.572	.374	13.61	.949	2
M.....	210.	40.865	11.5	11.5	.507	.281	18.26	.950	3
N.....	168.	36.	11.46	11.46	.482	.318	14.66	.820	3
O.....	300.5	30.5	9.583	9.583	.428	.314	31.35	.757	1
P.....	439.17	60.2	28.	28.	.795	.465	11.96	.984	2
Q.....	245.	23.52	6.687	6.687	.41	.284	36.64	.77	1
R.....	440.	56.	28.625	28.625	.797	.5112	9.083	.984	2
S.....	440.	56.	28.	28.	.79	.51	9.5	.984	2
T.....	252.	50.	12.583	12.583	.719	.2516	17.80	.955	3
U.....	250.17	41.67	14.852	14.852	.461	.356	16.83	.734	2
V.....	160.	25.	8.25	8.25	.886	.33	7.27	.980	2
W.....	160.	16.063	5.3	5.3	.370	.331	27.55	.720	1
X.....	147.	16.375	4.3	4.3	.447	.2626	33.95	.724	1
Y.....	99.5	12.5	3.81	3.81	.337	.305	24.67	.800	1
Z.....	200.	25.5	7.25	7.25	.47	.2843	26.76	.80	1
A.....	50.	9.5	2.5	2.5	.5208	.263	20.	.725	2
B.....	385.	53.	24.5	16.81	.7842	.4623	14.75	.967	2
C.....	210.	45.1	13.	13.	.578	.288	16.15	Less than standard	3

For spectacle frame ships H^1 = height above base line where tangent line to lower side of spectacle fin would cut after perpendicular. H^1 in all cases equals vertical height of rise of after body angle directing flow of water to propeller. For spectacle frame ships, K should be taken from curve C_3, Plate 74.



made than the former, as the type seems to depend very considerably on the ratio of draught to beam.

Generally speaking, the majority of vessels whose ratio of draught to beam is equal to or greater than the standard given on Plate 71 for types 1 and 2 vessels are of type 2, those from a little below the standard given for type 3 up to the standard given for types 1 and 2 are of type 1, while type 3 range from the standard given for type 3 and below it.

In submitting the results of my years of labor to the members of this Society I wish to express to the Society my appreciation of the many kind words which have been spoken to me concerning my work and to assure the members that the pleasure I experience on hearing them adds greatly to the pleasure I have found in prosecuting the investigation, a task which, while work of the hardest kind, has been of the most intense interest.

THE CHAIRMAN:—When I was young, about all we knew concerning propellers was that their ways were very peculiar. Admiral Dyson has become an international authority on these peculiarities and has illustrated them to an extent that sometimes it has seemed doubtful if it were really possible to do—the actual performance of propellers, and the application of this actual performance to the matter of design in a way that no other authorities, as far as I am aware, have succeeded in doing.

Before we adjourn, I wish to say that the Society thanks Admiral Dyson for his very elaborate paper, which has involved an immense amount of research, as is apparent to all who read it.

I want again to announce for the benefit of those not here this morning that there will be an excursion to the Federal Shipbuilding Company, at Kearney, N. J., tomorrow. The steamer leaves the Pennsylvania Railroad Pier at Cortlandt Street at nine o'clock sharp. Tickets may be obtained at Room 604.

I think we may congratulate the Society that in this meeting we have had most interesting sets of discussions on the various papers presented that have been placed before the Society, and I have no doubt that the members will have great pleasure when they receive their printed proceedings in reading over these discussions with the added points that may be made through written communications.

The session now stands adjourned.

TWENTY-EIGHTH ANNUAL BANQUET

OF

THE SOCIETY OF NAVAL ARCHITECTS AND MARINE ENGINEERS

HELD AT THE

WALDORF-ASTORIA HOTEL, NEW YORK, N. Y.

SATURDAY EVENING, NOVEMBER 13, 1920

REPORT OF SPEECHES AT THE TWENTY-EIGHTH ANNUAL DINNER OF THE SOCIETY OF NAVAL ARCHITECTS AND MARINE ENGINEERS, HELD AT THE WALDORF-ASTORIA, NEW YORK, N. Y., NOVEMBER 13, 1920.

After the service of the dinner, Mr. Homer L. Ferguson, acting president, called the company to order and, as toastmaster of the evening, said:—"Gentlemen, as is usual on occasions like this, our first toast of the evening is 'To the President of the United States,' which I ask you to give standing."

The toast was given standing, and the orchestra played "The Star Spangled Banner," in which the company joined in singing.

THE TOASTMASTER:—Gentlemen, we had hoped to have here this evening General Bullard and other representatives of the Army, and also several naval officers and admirals in addition to Admiral Glennon, who we are glad to say is with us. Our second toast on occasions like this is "To the Army and Navy of the United States."

This Society is closely affiliated with the Navy and always has been. In 1892 a band of officers, of whom your president, Admiral Capps, was one, started the Society with the assistance of an equal number of civilians. Our work has always been identified with that of the Navy. The Army has also been friendly with us and has always been represented here, and as I have not any speech to make tonight, and nothing in particular to say, I hope you will pardon me for taking a minute of your time in referring to the Army and Navy. Having been brought up in the service, I suppose I am somewhat partial to the Navy. Yesterday was Armistice Day. Two years ago yesterday the country resounded with the huzzas of the people for our Army and Navy. Today you hear little about it. The cry of militarists is in the land more or less, and yet during the great war we had a phenomenon in this country that has never before been seen here, and that is that the work of our Regular Army and our Regular Navy was so well done that there was not one note of dissatisfaction over all this country, nor a demand to return to this country any of the officers in command. (Applause.) That is a wonderful record, and whatever may be said to the contrary, General Pershing and Admiral Sims had their own way in France and in England. Two million of our soldiers went over on ships provided by us and by our Allies without the loss of a single man, and our soldier boys showed by their performance at the front that they were well trained and demonstrated to everybody's satisfaction that West Point was justified a thousand times in the character of the men that it sends out to train your boys and mine. (Applause.)

But at this time, when we are so taken up with affairs of business and in our effort to make the world perfectly good and beautiful and true, it is just as well to remember that when we needed them the officers and men of the Army and Navy were on the job, did their work to the Queen's taste, and came home victorious, claiming nothing and getting less in many cases. (Applause.)

I think that on the anniversary of Armistice Day it is up to us who have profited

by the splendid work done by these splendid men to drink to their health, and therefore, gentlemen, I give you "The Army and Navy of the United States." (Loud applause.)

The entire company then stood and drank to the toast.

THE TOASTMASTER:—Now, gentlemen, we have only two speakers for this evening, and you will not be detained long. My chief regret this evening is that I am in the talkative class. I do not feel justified in becoming talkative, as some of my friends have this evening, much as I would like to, because it is up to the toastmaster, usually, to keep his head, and it takes mighty little to turn one's head nowadays.

But I do want to congratulate you all upon being in the shipping and shipbuilding business. We have noticed recently in Virginia that there is no trouble at all in getting perfectly husky and fine mechanics to work on repair work—overtime, holidays, and nights particularly. In fact, one of the chief jobs of our watching force is to prevent the setting up of small illicit bars around the plant, at which drinks are sold to the workmen at 50 cents each.

The other day, when an Italian ship entered the dock, I noticed a man come off the ship—he went aboard as if in a great hurry and came ashore as if miserably deformed. (Laughter.)

But the shipbuilding and shipowning business is quite popular now, and those of us who are located on the seacoast look for a tremendous increase in population; in fact, I think it must have something to do with the shortage of housing.

We are all interested in the American Merchant Marine—we have heard a great deal about it, and we will hear still more about it before we have one, a real one. (Laughter.) But we hope that the youngest members present will live to see the time when there will be no doubt about its existence and its continuance.

The unfortunate part of it is that some of us insist that it shall be American—of course, we have always been provided with the other thing—and we have tonight, as one of our principal speakers, a gentleman who for years has represented the Fourth Pennsylvania District in Congress, and is the second senior member of the House Committee on Merchant Marine, and who, as you all know, has done splendid work in connection with the congressional investigations of that subject. He was instrumental in the presentation of the Merchant Marine Bill that was passed at the last session of Congress, and in particular his work in the investigation of insurance matters was the best along that line that has ever been done in any Congress. His interest is just as intense as yours and mine; he has always been our friend.

I regret that at this particular time there should be so much publicity of a certain kind in regard to shipping. Of course, when Noah outfitted the ark for a journey of forty days and nights, I daresay that the steward demanded something from the outfitters. (Applause.) I believe that the custom has continued from that time to this; but I do think it is a pity that so much publicity should be given to certain phases of the case, so that the public obtain an impression which is distorted. Heaven knows that everyone connected with the new merchant marine has enough to answer for without over-emphasizing any particular point.

However, we are very glad to have with us this evening the gentleman who will now address you, and whom we are very glad to have with us. We will now drink to the toast, "The American Merchant Marine."

I take great pleasure in presenting the Hon. George W. Edmonds, who will respond to the toast, "The American Merchant Marine."

ADDRESS BY HON. GEORGE W. EDMONDS.

Mr. Toastmaster and gentlemen, I am talking a little under difficulty tonight, but I will try to make you hear me. During the late unpleasantness, about ten days ago, I unfortunately paraded in a rainstorm. It was at night, and after the parade was over we had a little jubilee. I did not get home for three or four hours, and having wet feet, I contracted a cold. My interest in the merchant marine of this country is so great that when I received Admiral Taylor's letter, asking me to speak to you on this occasion, I thought possibly I would be able to get rid of the cold and come over and do my best.

I am not a lawyer. I am not an orator. I want to talk to you in just plain words the way I feel about the merchant marine. I know you are all interested in that subject, and it pleases me to see such a large gathering of men here tonight, who, from your looks, are, I know, going to assist us, the Congress of the United States, in putting the American merchant marine upon the sea. One of the difficulties, we were told in Washington, was that we did not have shipping brains in this country. I was inclined to believe that was true, but I will go away from here tonight a little in doubt after seeing you gentlemen.

During my earlier days it was one of my greatest pleasures to hear the late William Wagner, who founded the Wagner Free Institute in Philadelphia, speak to us on the business exploits of Stephen Girard. William Wagner was one of Stephen Girard's supercargoes. In those days they did not have telephones, automobiles or cables, and the captain of a ship was virtually the king of the ship, the supercargo was the business man, and William Wagner traveled from one end of the world to the other buying and selling cargoes for Stephen Girard. His stories were very interesting. At that time I was so enthusiastic over the subject of the merchant marine that I felt if someone came along and showed me the way, I would join a ship myself. However, in doing that I might not have made a very good sailor and might have spoiled a poor congressman. I have not been quite as enthusiastic over being a congressman as I was over the prospect of being a sailor.

At the beginning of the nineteenth century the ships of the United States did virtually the largest marine-carryinging business of the world, and at the opening of the twentieth century we did less than 10 per cent of our own business. It seemed peculiar that such should be the case, and of course, when it came to preparing a bill authorizing the Government to operate the ships built on account of the war, it was necessary for us to look into the reason. I am going to enter more into that tonight, probably, than you would wish.

Of course, during that time, and particularly after the Civil War, the developments of the country internally were so great that nearly all the money we had and all the money we could borrow went into local development inside of the country and were not used for developments on the sea. That brought about a lack of investment in shipping. It brought about a lack of interest in shipping. But the trouble really with our shipping was occasioned when Congress agreed to make treaties with different countries by which foreign ships had an equal advantage with our own ships in this

country. You might say that we ought to be able to handle ships as cheaply as another country, and yet with a tariff to build up the manufactures of the country, the same kind of protection would necessarily have to be given to the shipping in order to build up shipping. Congress removed that protection, and the result was there was no financial attraction for Americans to go into the shipping business.

The war brought one particular thing to our attention—that if we expected to remain an international power we could not do so unless we had a merchant marine of our own. We could not depend on other powers to do our carrying if we expected to send our merchandise to different parts of the world during a war. We realized that it was necessary to have ships of our own.

In 1906 there was a very thorough investigation made by Congress, and the committee in charge of that investigation came back with the recommendation of a subsidy. They did not see their way clear to put on preferential tariffs, and the result was that, as the interior congressmen would not stand for the subsidy, the bill was lost.

We have made our mistakes. Your chairman has spoken about the errors we have made in the past—I was talking over some of them with him tonight. I do not feel like entering into that. I have no doubt that anyone who would have endeavored to build up a large merchant marine would have made mistakes. Some of our mistakes, however, are not errors—they are international wrongs that ought to be corrected.

I know it is one of our popular games just now to find fault with everybody's business on the score of grafting and profiteering except your own—of course, your own never has any grafting or profiteering in it—so I will not enter very much into that discussion.

I want to go a little more into the Jones Bill and the reasons back of that bill, and the effects of the bill if put into working shape and practical operation.

I was very much disappointed when the President refused to abrogate our commercial treaties. When the LaFollette Bill was before Congress I believe the Department of State took the stand that Congress had no right to abrogate treaties without an agreement with the department. The President then decided that they had a right to abrogate treaties. The Jones Bill was passed with a full understanding of what was in the bill. The President had an opportunity to veto it, but he did not do so. He made it the law of the land and then refused to carry it out.

That particular section of the Jones Bill is going to stand in a very peculiar way and in a very successful way in keeping these ships operating. I have had opportunity recently to look into what would really happen if we abrogated these treaties and what would come to our ships in the way of a subsidy if the preferential duty clause in the Underwood Tariff Bill was put into effect, and remember the Underwood Tariff Bill is the lowest tariff bill we have had for years.

Take macaroni—we imported 75,000 tons of macaroni into this country in the year 1914. I do not know what we are doing now, because I have not later statistics of the quantity brought in. The duty on that quantity of macaroni was $1,500,000. If the 10 per cent preferential clause went into effect, the American ships carrying it would have a differential in their favor of $550,000, besides the special discount of $75,000 which is given by the sixth clause of the same section of the tariff bill for goods brought in in American ships; in other words, there would be a total preference on 75,000 tons of macaroni of $625,000 in favor of American ships.

Take diamonds. In 1918—I have later data on diamonds—we imported

$13,000,000 worth of diamonds in the rough. The duty was 10 per cent, and additional duty would be an additional 10 per cent, which would mean $1,300,000. Mind you, gentlemen, the diamonds could probably be brought in in a small grip. The 5 per cent off the duty gives the American ships an additional $65,000, a total of $1,365,000 in favor of the goods being brought in in American ships. You must recognize the fact that with this clause in operation our shipowners would be able to get better freight rates and would be able to command return cargoes that we could not otherwise get. The question of the difference in the wages, in the difference of the cost of operation, does not mean anything when compared with these figures.

We also brought in $13,800,000 of cut, but not set, diamonds, carrying 20 per cent duty. The additional duty on them would be $1,380,000, and the 5 per cent special discount, $69,000, making a total of $1,449,000 in favor of American ships, and this, mind you, would be but a very small portion of the cargo of the ship.

Take broad silks from Japan. In 1918 we imported 2,000,000 pounds, that is, 1,000 tons. The duty was $5,100,000; the additional revenue, if brought in foreign ships would be $1,150,000, and with the 5 per cent discount, $250,000, a total of $1,400,000 in favor of American ships on 1,000 tons of freight.

Take Axminster carpets. We brought in 400,000 yards in 1918, valued at $2,150,000 with 50 per cent duty. In that case 10 per cent extra if carried in foreign ships would be $215,000, and the discount of 5 per cent would amount to $53,750, or an amount in favor of American ships of $268,750, and mind you, gentlemen, this is without taking anything out of our pockets.

In the writing of the Jones Bill, the question of a subsidy was not considered, because it was believed in this section there is a sufficient advantage to keep our ships at sea through their ability to command a higher freight rate, because when they arrived at a foreign port it would be an advantage to the foreign shipper to send his return cargo in the American ship rather than to pay the additional duty if he sent it in a foreign ship.

There is another benefit in the Jones Bill, and that is that the import and export rates provided in the railroad legislation should be only applicable to American ships—that if foreign ships come into our ports they would have to pay the local railroad rates. I can cite one instance of that—a rate made before the last advance in freight rates. I have no record of the new rate, but wool from Australia was imported to and carried a rate from San Francisco to Boston of $24.50 or $25. Local wool grown in this country, in California, carried a rate of $49.50 to Boston, a difference between the foreign wool and American wool of about $24. This means, when placed into operation, a difference between a foreign ship and an American ship of $24 a ton. These are some of the means by which we expect to keep our American merchant marine on the sea. We have the matter well thought out, and there is no mistake in our judgment, in my opinion. We made up our minds that this country must have a merchant marine, and we propose, if it is possible by legislation or by some other means, to keep these ships built by the Government at such great expense on the sea. (Applause.)

In 1915 I spent seven months in the Orient. I saw the old New York come into the Olongapo Navy Yard in the Philippine Islands with the American flag flying. I saw a few boats of the Pacific Mail Steamship Line, then going out of business. The last trip of the Manchuria was made while I was in Japan. Outside of this I saw very few American flags upon the ocean anywhere—U. S. transports, yes; Pacific Mail Steam-

ship Line going out of business, yes; and some of the Robert Dollar boats, maybe. You never knew where they were going; they seemed to change around here and there. I do not know whether Robert Dollar is here tonight, but that is true, and he knows it. Some boats with a German flag I saw—they were interned. But the American flag, outside of the coastwise ships and the United States fleet and in the Philippines, was very little in evidence. I felt sad to think I had worked for four or five years on the Merchant Marine Committee and had not been able to assist in evolving some scheme to keep the American flag on the sea.

We hope this bill is going to keep the American flag on the sea (applause), but it is not going to keep it there unless you gentlemen leave this room tonight and make up your minds you will help. It is not going to be kept on the sea by exorbitant charges against your ships—by foolish business methods, by subsidiary shipping companies that provide supplies to the ships, by subsidiary repair companies that get the rake-off of the repair yards. It is going to be kept on the sea only if you gentlemen will make up your minds to see that all these wrongs are corrected and to see the ships of the United States treated as fairly and as honestly and as rightly as foreign ships are treated. (Applause.)

I will go a little further into the Jones Bill. One part of the bill is arranged for rebating taxes, so that you can build ships, and I presume that is particularly of interest to you in your line of work. You men mostly are in the shipbuilding business and want to build ships. This rebating of taxes will give you business. The establishment of the American Bureau of Shipping, I believe, is now successfully launched, and men in the shipping and shipbuilding business must make use of the American Bureau of Shipping. You cannot make these ships operate with foreign tools. You must operate these ships with your own tools. You have enough ships on which to make a good start, and we are trying to give you the tools for more enlargement, so it is up to you to use them.

The establishment of the preferred mortgage was quite a step in advance. I do not suppose my ship-repair friends will like it, but we thought it was advisable to give two statuses to the mortgage, one the old status and the second one the preferred status, so in event a man wanted to borrow money and could not find anyone to lend on one kind of mortgage, he could borrow on the other.

The protection of the coastwise trade has always been American and always will be continued, I hope. The postal subventions which have been hardly called upon by the Shipping Board is another tool that can be made use of to keep the ships operating. The assistance of the army engineers in developing ports will be a good thing.

The new section, which will prevent the giving of deferred rebates to any ship coming to this country and making the penalty the exclusion of the ships of such lines, is very valuable and will help us considerably. Further, it is required if for any reason the foreign shipowners band themselves together and give deferred rebates between foreign ports, if they do not give the American ship in that same trade the same opportunity to join in the deferred rebate arrangement their ships will be excluded from our ports also. (Applause.)

One thing that has particularly puzzled me is why it is—and this is true of New York—why it is that we have such poor facilities at our ports. All of the ports that expect to be selected as terminals for this merchant marine had better look to their port facilities. I have been amazed to find the tremendous costs of the handling of

freight in the port of New York, and I think that is true, to a greater or less extent, of all of our ports. I believe today that the two model ports in this country are New Orleans and Seattle. I am not speaking for my city, Philadelphia, for I think we need a lot of changes, but at the same time the improvement of these ports so as to conduct the loading of these ships rapidly and efficiently is going to assist materially in making the ships profitable.

Another thing—and I have been working all day on this—is the use of American facilities. We found in the insurance investigation that when an Englishman came over here and brought any large bulk of goods that he insisted that they should be insured in English insurance companies, that they should be financed by English banks and carried as far as possible by English ships.

I call your attention to the arrangement made in New Orleans a short time ago for the forwarding of cotton. In every way they endeavored to assist the different branches of the trade in their country. Unless you can get that spirit among you, gentlemen, you are never going to make the merchant marine a practical success. It may be, possibly on account of the fact that your different facilities are in their infancy, that you may have to pay a little bit more for insurance; it may be possible you may not get such an advantageous policy, because the English people, who handle most of the marine insurance of the world, have other places, other countries to draw on. They do an international business. They do a business all over the world. They can make your rates low and some other country's high in order to make up the loss on the rates here.

You cannot get these things without fighting, and we have got to fight to retain our commerce, exactly the same as they have had to fight to build up theirs. You are now where they started some hundred and fifty years ago, and unless you are prepared to make some sacrifice to build this merchant marine you are not going to build it up. You must use your banks wherever possible, you must use your ships wherever possible, and you must use your own insurance companies wherever possible if you expect to make success of the American merchant marine. These three things are the vital parts of the business. The economies that can be effected in repairs, the economies that can be effected in supplies, are all matters of pure business.

I was reminded in a talk this evening with the toastmaster in regard to the old way of paying captains. Some time ago I was in business in Philadelphia, and I sold coal along the Delaware River to the captains of steamers. One day a man came to my office—he was a sort of an agent—and bought 163 tons of coal in a boat, and right in front of me he changed the bill of lading to 400 tons and took the boatload of coal up to a steamer.

I said:—"What are you doing?"

He replied, "The captain has got to be taken care of and the engineer and some other fellows."

"You don't give them all of it, do you?" I asked.

"No; I keep most of it myself," was the reply.

That is true, wherever there is any graft of that kind. The other fellow gets his share, and in an investigation I made in the last six months I found that the other fellow, where this practice exists, does get his share. I found one ship that loaded about $4,000 worth of groceries and supplies which could be purchased at a wholesale grocery

house for about $1,600, and it was very evident that the captain did not get the $2,400 in that case. Certainly, somebody got a nice profit out of it.

Let me tell you what your English competitors do and have done for some years. They have agents. These agents may be agents for one hundred ships or fifty ships, and they contract, at every port where they think these ships are going to touch, for coal for their ships. Neither the captain nor the engineer has anything to do with it. I remember when they first started this method of doing business. The engineers and the captains kicked, and were fired. Only two or three were fired, and the coal was good afterwards, the voyages were made just as speedily, and that practice is being pursued today. The steamship owners call for bids for the supplying of coal, and certain companies bid on the coal supply at such and such a price, and the captain no longer has anything to do with it. Nobody has anything to do with it but the officials of the company who call for the bids and the concerns which make the successful bids. The bills are sent forward to the company. There is no necessity for doing any trick work in connection with the weight of the coal, because the coal dealer who would do this would be simply a plain thief. He has no reason for doing it he is not forced to do it, and he does not have to do it. I know of no man in Philadelphia who had contracts of this kind who would do it, an I am very sure there are many men in the coal business in New York who would not do it.

I want to call your attention to the way our English friends feel about our shipping. This is a statement by Mr. J. C. Gould, a member of Parliament, writing in the *Pall Mall Gazette*. He is a British shipowner and operator. He has made a personal investigation of the shipping situation in the United States, and among other things he says:—

"It is difficult for an Englishman to criticise American shipping without an accusation of bias or prejudice being brought forward, but the fact remains that among the first to condemn the present administration are Americans of influence, judgment and discernment, who realize that the whole growth of American shipping has been due to emergency methods, has been carried on regardless of expense, has been placed in the hands of people lacking in knowledge and experience of ship management, and is based upon a system of costs so high as to make economical operation an impossibility. Then again, the speed and haste with which so many of the vessels were built has resulted in a great number of vessels which do not measure up to anything like the standard of foreign tonnage, and the navigation laws are diametrically opposed to sane administration of shipping, and, moreover, the fact that the majority of the ships are owned by a government department does not make for economy of management, but instead tends toward delays of operation which prove very costly in the long run."

Later on he says:—

"Stores, equipment, management and incidentals, including depreciation and repairs, which must of necessity be considerably higher than on British ships, add to the total of the costs, so that on analysis one finds that the operative costs are anywhere between two and one-half and three and one-half times the cost of operation of a British vessel. It is beyond doubt a fact that British vessels will be making money at very much lower freights than prevail today, while American vessels will be losing heavily at the same time.

"Personally, I do not think that the American people will agree to subsidize a mercantile marine. Evidence is forthcoming here on every hand that if American shipping is to have a chance of making itself a permanent institution its management

in all its phases must first of all pass into the hands of experienced and reliable private individuals; secondly, it must be reorganized by men of proved ability and experience; thirdly, sane shipping laws must be enacted; and fourthly, construction and operation costs shall be reduced to a level of comparison with Great Britain."

He states four reasons, and I agree with him in all excepting the construction cost. My answer to that is the statement of Mr. Harriman, who has been building ships at the Chester Shipbuilding Company, near Philadelphia. He thinks we can build as cheaply today as the British, and I agree with him. Therefore, let us hope that question is removed for the present.

It is undoubtedly true that, unless we use all the brains and all the ability that we can find, we are going to have a hard time when we return to a competitive basis to keep these ships operating, but I do believe that with the Jones Bill and its beneficial features we can keep these ships on the sea and gradually remove the obstacles and eventually find that we can compete with other maritime nations.

I am not so sure that the navigation laws are going to be very troublesome. A committee composed of Mr. Franklin, Mr. Taylor and old Captain Dearborn came down to Washington, and we asked them to tell us what was the matter with the navigation laws. After a long study they returned us a complaint about some regulations in the carrying out of the navigation laws, and that is about as far as we could see that there was anything wrong with the navigation laws, but if there is anything wrong with them, I am certain that the Merchant Marine Committee of the House would be glad to know of it. If there is anything that we can do to correct them, anything that is unsatisfactory and can be corrected, I know we will be glad to do it.

But do not forget this—these special privileges we are giving you under the Jones Bill are intended to keep up an American standard among the sailors of this nation and to make the business of the sea an attractive one to the young men of this country. (Applause.) Now if you are going to try to drive the American sailor back to the status of the East Indian sailor, the Lascar or the Malay, we will object to it. We want our men on the sea to be properly taken care of, properly housed with proper rooms. We do not want them to be coddled and we do not want them to be undisciplined. I understand there is a good deal of trouble about the discipline on the ships. I think you will find with the passing away of our labor troubles on land that the discipline on the ships will improve, and that the troubles which are incident to all labor matters at the present time will end.

With all the arrangements which have been perfected to assist, there is no reason, if you play the game fair—and I am putting it up to you men tonight, that we expect you to play the game fair—there is no reason why you should not find the Stars and Stripes on every one of the larger seas of this great world. (Applause.)

THE TOASTMASTER:—We are very much indebted to Mr. Edmonds for his very clear statement of the merchant marine bill which was passed in the last Congress.

I have been asked by a telegram from Admiral Capps, the president of this Society, to express to you his regret at not being here tonight, as he was ordered to the west coast on official business by the Navy Department and will be gone for several weeks. He asked me to say to you particularly that he was sorry not to be here or at the meetings.

In connection with our entertainment tonight, I feel that our thanks are particularly due to the Entertainment Committee of the Society, headed by Mr. C. A. McAllister, chairman—it seems odd to call him anything but "Mac"—and associated

with him are Mr. J. Howland Gardner, Mr. W. J. Parslow, Mr. W. W. Smith, Mr. H. R. Sutphen, Mr. F. L. DuBosque, Mr. F. G. Coburn, Mr. H. H. Raymond, and Mr. H. N. Fletcher. They are responsible for the arrangements made for your entertainment this evening, and I am sure you will wish me to thank them in your behalf.

Our next speaker is from the good old shipbuilding city of Brooklyn. He was a stranger to me until this evening, but now we are very friendly. His folks went from Ireland to Scotland and then came over here, and my folks in the meantime were engaged in dodging the authorities in Scotland to go to Ireland and then came over here. So we are both Scotch-Irish, and we have a common bond of sympathy in that we have been mixed up in one way or another in every row that has happened for a long time where machinery and modern implements of war were used.

The gentleman who will address us comes of a shipbuilding family, but he tells me he gave up shipbuilding to go to Congress and recently was caught in the enormous political landslide that happened. He has been elected a Justice of the Supreme Court of the State of New York.

Down our way we were surprised about the election. A friend of mine, a former Democrat, living in New York, and who is here tonight, said he didn't know what had happened, but apparently there was a lot of discontent in the United States; a discontent that is probably quite natural under the circumstances and which may be repeated in some other direction later on—you cannot tell.

Whoever is to straighten out the mess that we are in certainly has a fair-sized job and is to be sympathized with from the start to the finish. I think the difference between some of us who call ourselves Democrats in the South, and some others in another section of the country is not perhaps as great as might be assumed. Those of us who are engaged in industry, and who in recent years have watched the tremendous concentration of power in certain places, or where men at the head of great unions deal with men at the head of great aggregations of capital or heads of governmental departments—I am sure that the thinking men in industry feel that centralization has rather gone the limit so far as we are concerned, and that perhaps it was just as well we looked over ancient democratic notions, which are that a man shall govern his own household and attend to his own business, and settle his rows with his own neighbors, run his own local government, and have just as much a centralized form of authority as is necessary and no more. (Applause.) It may be true that, after all, the ancient Democrats were not as far wrong as we may think in insisting that a reasonable degree of state and local rights is necessary to the welfare of a great republic.

I am not going to make a political speech, but I just want to say that, although we have no kick whatever on the election (laughter)—any more than you had four years ago—that any industrial concentration of power, so that the contact between employer and employee is removed from the particular establishment and carried off to the seat of government, is an un-American conception. (Applause.) We must show that, no matter how much friends from abroad shall insist that the closed shop is the only method of operating a business, there are a lot of us in the United States who do not think so.

Gentlemen, the next speaker has not been assigned a subject.

It is a great pleasure to have Judge MacCrate here with us this evening. He will respond to the toast, "The Past, the Present and the Future." It gives me great pleasure to present him to you—The Honorable John MacCrate!

ADDRESS BY HON. JOHN MacCRATE.

Mr. Toastmaster and gentlemen, of course the toastmaster did not tell you that the reason I have not any topic assigned to me is because I am substituting for a great man, and I am here because somebody else is in Cleveland or some other place. If the junior senator from the State of New York, who has always been a friend of the shipping industry in this country and one of the best equipped men in the Senate of the United States on that subject—if William M. Calder were in New York tonight I would be in Brooklyn in my bed. (Laughter.)

But I am perfectly willing to go to any place where I may meet American men interested in shipping and tell them of the work of men like my friend Edmonds in the House, and Calder in the Senate, business men, who do not profess to have the gift of gab, but who do have the saving sense called common sense, and who, in their committees in Congress, make people show them why things should be done and what should be done.

Mr. Edmonds is chairman of the Claims Committee, of which I am a member in Congress. If you can get five cents out of Uncle Sam with Edmonds in charge of the Claims Committee, you have got to invent some way that Raffles or Jimmy Valentine did not know anything about. Edmonds protects the Claims Committee and protects the Treasury of the United States, and as chairman of the Claims Committee he is most obdurate, only to go over into the Merchant Marine Committee and do anything that the shipbuilders and others ask him to do.

I have been trying to get a claim for $26 through the Claims Committee, and by the way in which that has been obstructed you would think I was trying to take the Treasury back to Brooklyn. Now that I have fourteen years of occupation staring me in the face at a fairly reasonable income, he can go to blazes and I will find somebody else to get the $26.

I found myself thinking, as some remarks were made about some people in the audience by the toastmaster, that Edmonds and myself are in bad odor here and that we are blamed for the present drought. We voted wet, but the majority were against us. (Laughter and applause.) If you think that the Sahara Desert is dry, go down to Congress when they are voting on any prohibition measure. Some people say they wondered how Volstead ever could win. Volstead told us, when he introduced the bill:—"I will be beaten in the next Republican primary in Minnesota and I will be beaten by dry." I said:—"What kind of a country do you have out there?" It is wonderful how your conceptions change as you grow older. I have not grown so much older, but my conceptions have changed.

I used to read the American Statesmanship Series of our congressmen and senators. Among them I read the lives of Henry Clay, Daniel Webster and Charles Sumner, and I noticed they wore a long undertaker's coat, a silk hat, and carried a cane. Well, since I have been in Congress I have not seen any undertakers' coats down there. Everybody wears a business coat, and if you wear a silk hat it is only when you go to a wedding, and in the districts from which Mr. Edmonds and I come they would not allow you to use a cane.

Your conception of Congress changes. About 18,000 bills were introduced in the last Congress. You would think that Congress was a big drug store and the President was a great physician, and every time the country had an ache or pain all you had to do was to get a presidential prescription, send it in to Congress, and then Congress would pull down some bottles of dope and fill it into a sugar-coated pill and give it to the country, and that would cure the country's condition. Congress cannot perform any such feat, and you will never cure America in any such way—simply by legislation. The worst abuses that will come to this country will come through legislative halls, and just as soon as our people turn their eyes away from legislative remedies and give their attention to what has been always fundamentally American, just so soon will we solve our problems, and not until then.

As to these 18,000 bills, most of them, if they were enacted into law, would completely change the form of our government and the structure of things we believe in. They were absolutely out of harmony with fundamental American conceptions of government. I wish I had brought with me—I intended to do so—some six or seven postal cards I received from children in my district, who cannot be more than nine years of age, judging from the handwriting on the postal cards. We get many postal cards in Congress. Somebody said to me:—"If you want to come back here, the way to do it is to write a letter to every one of your constituents telling them you have seeds and books to distribute, and if they want any you will send seeds or books to them." I sent a letter to my constituents telling them about the fine flower seeds and the various pamphlets and saying I was ready to send them to anyone requesting them, and I got back one postal card reading like this: "Dear John:—I do not want any of your seeds and I don't want any of your books, but when do I get my beer back again?" (Laughter and applause.) If I could only answer that, he would never know. I would go out and make money on that knowledge.

But I have six or seven postal cards from children, not in response to a letter of mine, but they came to me, and one read this way:—"Dear Sir: Please send me a walking doll and a carriage and everything that goes with it." Another read:—"Please send me a painting set." Another read:—"Please send me a cracker-jack book." Another said:—"Please send me a walking doll, a painting set and a cracker-jack book." Men, do you believe it possible that a child seven, eight or nine years old wrote those postal cards—that they were written by children of that age? How do they know who their congressman is, and then, if they know, how do they know where his place of doing business is?

There is somewhere in this country propaganda among the children of the country to teach them that the government is a great supply house where, if you want anything, they will furnish it to you. All you have to do is to ask for it and it will be given to you, and if it is not given to you, then you should be, as the child said in one of these postal cards, "mad" at the Government and ought not to have any liking for it. Certainly childhood does not write requesting that out of the great toy house of government, things be given to them unless someone tells them to do so. But that conception is not confined to childhood. The conception that government is something to live on rather than something to live under is quite prevalent and common, and a call for legislation for this, and legislation for that, and legislation for the other thing demonstrates clearly that there is a growing belief that the creature called a government is in fact a creator. Americans must everlastingly keep in mind that government is

the creature, and government is not the creator, and if we have a great government, it is only because we have a great people and that people has created the Government. The people have not relied on the Government in the past, but the Government has relied on the people, and if we change the emphasis from the people to the Government, we will change not only the character of government but the character of our people as well.

Our Americans have never been a leaning, clinging race, but they will become such if the constant emphasis on legislation as a cure-all is continued.

Do you know, men, in contemplating some of the things asked for by law at first blush they seem to be worthwhile, but after you have examined into them you find that they are dangerous. During the anthracite coal trouble all the people of the country clamored for a law to curb the miners. Somebody said:—"Jail them, jail them, jail them," never stopping to think that if you can extend the arm of the law out to jail the miner, some day you will extend the arm of the law out to jail the mine owner.

You cannot emphasize in this republic the obligations, responsibilities or liabilities of one group one day, without having a corresponding agitation for obligations, responsibilities, and liabilities for another group another day. Today you have the same problem—people crying aloud for industrial courts to curb the workingman, as it is said, and what do you see as the result? You see in the morning papers that in Kansas they have haled the manufacturers before them, so as to investigate why there has been a lock-out and the workers laid off. The danger is, if we call on legislation for a specific purpose at a given time, that specific purpose will be lost to view, and general control will be substituted for special control at a given time. We face in this country at the present moment a danger of $4,000 clerks attempting to regulate industry and enterprise that requires brains worth $50,000 to $100,000 a year to guide.

If we look at the creature called the government and we say the government is perfect and can make no mistake, we have lost an American conception, because our forbears never believed in the total perfection of governmental agents. Neither did they believe in the total depravity of the people, but there is an emphasis now on the depravity of everybody else except those who want a special law passed. There was a law written thousands of years ago that cannot be abrogated and cannot be repealed by any legislative body, and that was:—"In the sweat of thy brow thou shalt eat thy bread," and just as soon an our American people realize, as did our forbears, that sweat on the brow means more than ink on the legislative page, we will solve our problems. (Applause.)

But the trouble is that many people want to count the sweat drops on your brow, but do not want to sweat themselves. Washington is filled with men and women who want to card index what everybody else is doing. Wherever you turn there is somebody with a scheme to save society by establishing a card index and putting in six assistants, with one chief at $10,000 a year. You cannot increase production by taking producers out of productive life and putting them at a desk in Washington or in any other capital in this country. We must keep our people producing without unnecessary interference from people who do not know anything about the business in which our people are engaged.

I have often wondered why it was that there was not a better understanding between

the men employed and the employers, but I went over to the place where your chairman was with his parents, as I understand it, into Glasgow, two years ago, and went to Dumbarton on the Clyde and saw my uncle, who worked for forty-seven years in McMillan's Shipyard, and another uncle who had been there thirty-six years. They were going on a general strike on the Monday following my arrival there, and I said:—"Uncle Jim, what are you striking for?" He replied:—"I will tell you, John, I do not like to leave, after being there forty-seven years, but the old governor, when I was an apprentice boy, used to get to the works at the same time that I did in the morning and he left a little bit after I left, but his grandson comes at nine in the morning with his limousine and leaves at three. So the boys begin to feel, if the governor works thirty-six hours a week, they ought to work only thirty-six hours a week, too."

Now that seems strange, at first blush, but may I suggest to you that perhaps we have lost sight of the fact that there is a human element involved, and that if the men employed notice the slacking up in the high places, they may slack up in their places. Uncle Joe Cannon put it perhaps in such a way that you will appreciate it as I did. A young man got up in Congress and began to talk about the hard times and the oppressive conditions under which our people lived. He said that the times were such that the men and women could not bear them any longer and that they must have relief. A remedy must be found for the intolerable conditions under which men and women existed.

I, as a young man, sat and listened and thought it was a pretty good speech until Uncle Joe, with his eighty-four years, got up in the front of the hall, looked at the young man and said:—"This is a patent bill; we are discussing patents. When I was a boy we did not have any electric lights, did not have any telegraphs, did not have any telephones, no steam railroads such as you have, no airplanes. We did not even have a big McCormick reaper. When we wanted to flail corn we went into the barn and took the flail in our hand and flailed it on the barn floor. Young man, if you have a hard time now, we must have had a hell of a hard time then." (Applause.)

Men, I simply suggest to you this—that the complaint among employers and among employees will, to a great extent, be stopped if we can only turn the thought of our people back along the path which we have come as a nation; if we can only teach them that macadam roads have not been the way Americans always traveled, and what we have now is not ours because we have made it but because we have inherited it from the toil and struggle and labor of others.

In this year, the three hundredth anniversary of the landing of the Pilgrims, it will be well for men who employ and men who are employed to throw their mind's eye back to the landing of the Pilgrim ship, and ask:—What do we bear compared with what they bore? What do we stand compared with what they stood, that this nation might be? Then there must always be a remembrance that the human being is in government as well as in industry.

A young man got up one day in Congress and said:—"What we want is men who will never falter, men who will never yield to temptation, men whom you can depend upon in any emergency to do the right thing." Uncle Joe Cannon arose and, standing right in front of him, he shook the ashes from his cigar, then leaned over and said—"Let us subpoena God Almighty." That was the answer, the complete answer—on earth, in industry, in government, human beings are going to do the work.

But we ought not to think too pessimistically about the future. Times have

changed and the demand that the action of citizens shall be controlled by law may be only temporary. I may tell you that in 1719, years before prohibition was thought of in this country, a man's nose was cut off in Turkey if he smoked. The world has changed since then, and perhaps we will have another swing, but we must swing back to that conception of duty where the individual desires to do merely what is right, regardless of law, that he does not consult the penal code or the criminal code to see what his conduct should be.

God help America, when employers or citizens look into the criminal code or the penal code to know what they may do, rather than consult their consciences as to what they should do with relation to their employees and fellow-men. We can never substitute the criminal code in the conduct of government or private affairs or controversies for conscience, which is not made in legislative walls but in the homes and schools and churches of the nation. (Applause.) Not only that, but there must be a total abandon to the good of the country.

Let me illustrate it. As I was going to Scotland two years ago I looked into one of those English cabooses and saw three good-looking young girls inside of the caboose, and I did what you would do if your wife were 3,000 miles away—I looked again, and I heard one say to the other:—"I will bet he is a Yankee." I asked:—"How do you know I am a Yankee?" The response was:—"Oh, come on in, we know one of our own kind 3,000 miles way." I went in and I saw on the lapels of the coats of the young women the insignia "U.S.A." They were United States nurse girls going on a day's trip to Edinburgh, having been furloughed for that purpose.

Across on the other seat was a Scotchman in his kilts and hat. The Scotchman spoke up and said:—"I have not saluted an officer since Armistice Day and I will never salute one as long as I live. I am sick of officers." One of the young girls said she came from the west, from Minnesota, that the west is God's country, and the further west you get the better America is. She declared that when she when back to Minnesota she would go as far west as she could. She loved the west. Then she looked at the Scotchman, and said:—"I would like to take a hat like that with me." He then said, "Gie me your name and address and when I get hame to Scotland I will send you this one." She said, "Oh no; you will want something to remember France by." He replied:—I have twa bullets in my right leg that I got in France, I canna' gie you one of those. I'll remember France."

You know how our Yankee girls do. She took out her writing pad, wrote her name and address on it, and said, "This is my name and address; now don't you forget." Sandy took the sheet of paper, put it in his cap and put the cap on his head again, and then the girl sitting next to me said:—"What did you Scotch boys think of our Yankee lads?" Sandy replied:—"Do you want me to tell you the whole truth?" She answered:—"Sure, go ahead; the sky is the limit." Sandy then said:—"I had been in France twa years before your fellows cam across at all. I was sick at hairt—I was shootin' men that never did anything to me. I was killin' poor devils that hadna done me any hairm, and I didna ken what I was shootin' them for, and I was sick. All my folks were back in Scotland, and I wanted to be wi' my ain folk again. Then, one day in North France, a train stopped, and suddenly a lot o' wild men, a lot o' wild rascals, rushed off the train and cam ower to me and shook me and said:—"Hello, Scotty; hello, Jock; lead us to the damned shooting gallery." (Loud applause.)

Not in that language, but with that spirit, Americans have gone forward. You

can see them step off the train, 3,000 miles and more away from home, some 6,000 miles away from home; you can see them step on foreign soil, and ask:—"Where is the shooting gallery?" You say there is death on the other side; yes, but it is for America. With a total abandon of all thought of self, with a laugh on their lips and a shout in the air, Americans have always gone forward to danger for the nation, and that spirit is amongst us yet—that spirit is not quenched; that spirit has not been stilled.

Next Monday morning, 23,000,000 children, boys and girls, will go into the schoolhouses of this country, and as you used to sing "My Country, 'Tis of Thee," so they will sing, "My Country, 'Tis of Thee," and as you read what Nathan Hale said:—"I regret I have but one life to give for my country," so they will read, "I regret I have but one life to give for my country," and the sentiments, the aspirations, the hopes and the ambitions that welled up in your childhood bosom will well up in their bosoms, because they, too, will feel that they want some day to do something great for this country of ours.

It means that the republic is safe, for, added to the 23,000,000 children now attending school, there are millions more who have never been spoiled by a life of luxury, never been spoiled by hatred of government, who seek no better form of government than a representative democracy such as ours, who seek nothing else but the chance or hope to go upward and onward that you have had, who will seek in their sons to have duplicated the opportunities which were apparent in the life of Lincoln, who will look at their boys and say:—"Some day we hope in America they can push up and do better than we have done, and be better than we are," who hope that the avenue up will be kept free from hurdles, or, if there be hurdles, that those hurdles will be common to all the citizenship, and that all will meet the same conditions. This keeping the way open for childhood is the hope of America, and if we can keep alive in the breasts of the 23,000,000 children of America the feeling and thought that here, regardless of parentage, regardless of ancestry, regardless of the country from which their parents came—that here in America the way up for childhood is open, we will be true to the ideals of America. It is our duty to build up a great representative democracy where each shall count as one, to keep forever hope alive in the human breast, so that childhood looking at our flag can say:—"There is no land like this; no flag like that flag; no spot in all the world where I can better develop the God-given qualities which I possess; no other spot where I can better be what I want to be; here character counts for more than gold; here not who are you, but what are you that counts." (Loud applause.)

The Toastmaster:—I am sure you will join with me in the warmest thanks to Judge MacCrate for his wonderful address.

Our program is now finished, and I desire to express my great appreciation of the splendid attention given tonight to the addresses. I wish you all good-night.

The company then dispersed.

Deaths, 1920

WILLIAM BOYD

MEMBER

William Boyd was born May 1, 1874. His education was obtained at the South Kensington School, London, and Technical College, Glasgow. Thereafter he served an apprenticeship and worked as hull draughtsman seven years on the Clyde, Scotland. Coming to this country he was employed as hull draughtsman by the New York Shipbuilding Company, Camden, N. J., eighteen months; Bath Iron Works, Bath, Me., ten years; Electric Boat Company, Groton, Conn., in charge of work, five years; T. E. Ferris, five months; and chief hull draftsman for the Groton Iron Works, Groton, Conn., at the time of his death.

Mr. Boyd died August 18, 1919.

CHARLES B. CALDER

MEMBER

Charles B. Calder, general manager of the Toledo Shipbuilding Company, Toledo, Ohio, who had followed the shipping business from boyhood and worked to the top from a minor position aboard ship, died January 22, 1920.

He was born January 15, 1853, in Antwerp, N. Y. A few years after Mr. Calder's birth, his father purchased a farm of 1,140 acres on Welles Island. Mr. Calder was brought up on the farm and went to school until he was thirteen years old. In 1866, when his father went into partnership with J. H. Crabb in a shingle mill venture, Mr. Calder went into the mill and ran the engine until November, 1872. At this time he shipped as a cook on a little schooner sailed by Edwin Cook. Later he went to Oswego, N. Y., and took out a license as second assistant engineer.

In the spring of 1873 he helped put the engine in the Utica, built by Robert Davis and Z. Wright. On July 1, 1873, he left the Utica, as his papers would not permit of him running her. He then shipped on the tug Sarah Daly and was on her until Novem-

ber 1, 1873, when he went back to the Utica, which was then running as a railroad ferry at Ogdensburg, N. Y. He sailed on her until the fall of 1874, when she was sold.

In the spring of 1875 he shipped on the Inter Ocean as second assistant engineer, sailing on her until 1879 when he shipped as chief engineer of the tug Niagara. In that season he went back as second engineer of the Inter Ocean. In 1880 he was made chief engineer of this vessel. In 1881 he shipped as chief engineer of the Escanaba, sailing on her until the winter of 1886, when he superintended the compounding of the Inter Ocean and the Argonaut. He then went to Cleveland to superintend the building of the Ira H. Owen. He had a small interest in this vessel and was her chief engineer until 1889, when he superintended the building of the Parks Foster. He was chief engineer of her until 1891, when he accepted the position of port engineer of the Menominee Transportation Company's steamers. In 1892 he was appointed port engineer of the Mutual Transit Company's steamers in conjunction with the above.

On March 1, 1894, he accepted the position of superintendent of the Dry Dock Engine Works, Detroit, holding this position until 1899, when he was appointed general superintendent of the Detroit Shipbuilding Company. He held this position until the fall of 1905, when he resigned. He was one of the incorporators of the Toledo Shipbuilding Company and was made general manager of that firm. In 1909 he was elected vice-president and general manager, which position he held until his death.

JONATHAN IRVIN CHAFFEE

MEMBER

In the death of Prof. J. Irvin Chaffee on December 3, 1919, the sciences of naval architecture and marine engineering suffered the loss of an educator who had been of marked service.

When the United States entered the World War, 80 per cent of the 192 graduates of Webb's Academy of Naval Architecture and Marine Engineering (now the Webb Institute of Naval Architecture) engaged in war work—in the Army and Navy, in ship design for the Navy, in the Emergency Fleet Corporation, in shipbuilding, and in special allied work. This brilliant record is due very largely to the ability, faithfulness and untiring labor of Professor Chaffee, who for twenty-four years taught mathematics and applied mechanics at the academy, and also during a considerable part of that period was its resident manager and the head of its faculty.

Jonathan Irvin Chaffee was born at Seekonk, Mass., on January 3, 1861. He was of old New England stock, the descendant in the ninth generation of Thomas Chaffee who in 1635 was a landowner and resident of Hingham, Mass. On the maternal side, Pro-

fessor Chaffee was descended from Capt. Benjamin Church, the hero of King Philip's war.

He received his early education in public and private schools at Providence, R. I., and entered Brown University in the class of 1883. In 1882, while still in his senior year there, he was appointed principal of one of the grammar schools of East Providence and established the first high-school course instituted there, maintaining meanwhile his standing at the university and receiving his degree in due course. Later, for a time, he followed advanced work in mathematics and physics at the University of Berlin, Germany, and at Johns Hopkins University. His subsequent service as an educator was varied and responsible—as principal of high schools or superintendent of schools at several places in New England, including Providence, R. I., and Stratford, Conn.

In 1895 he was appointed professor of mathematics at Webb's Academy. The work which confronted him there gave wide scope for the thoroughness which was his dominating characteristic. In accordance with the directions of the founder, William H. Webb, instruction was limited to the sciences of naval architecture and marine engineering, with only such preparatory studies in mathematics and mechanics as were required for these sciences. Therefore, Mr. Webb appointed originally but three instructors—in mathematics, naval architecture, and engineering. During the years since then, this faculty has grown to seven members, and the course has been lengthened to four years.

While the limitation as to the subjects taught prevented the academy from becoming of collegiate grade in the State of New York, this limitation had its compensation in the fact that, in a course four years long, these subjects could be handled with a completeness of detail which is not possible in the usual college curriculum. This thoroughness, although it involved much labor, gave Professor Chaffee keen pleasure. On the foundations, broad and deep, which he laid in fundamental branches, his colleagues were able to build with surety and success.

His marked executive ability led, in 1901, to his appointment, in addition to his professorial duties, to the post of resident manager of the entire institution established by Mr. Webb. The very considerable responsibility involved in this position is shown by the fact that the Webb Foundation is dual in its purpose—both as an academy for training young men and as a home for retired shipbuilders. As an executive in this work, Professor Chaffee was a man of force, swift in decision and action, and rigidly conscientious.

He was active in civic duties, serving as president of the Board of Trustees of the Union Hospital, Borough of the Bronx; as a member of the Bronx Board of Trade; as a member, and at one time president of the local School Board; and as a member of the Mayor's Committee during the administration of John Purroy Mitchel.

Summoned in the full tide of manhood, when seemingly there lay before him many fruitful and honored years, Professor Chaffee has left to his old students and his many friends the memory of a life marked by worthy achievement, by sterling character, by faithfulness and loyalty in all things.

HENRY CLAY FRICK

LIFE MEMBER

Henry Clay Frick was born December 19, 1849, in West Overland, Westmoreland County, Pa. His father, John W. Frick, was a farmer of Swiss ancestry, and his mother, Elizabeth Overholt Frick, a member of an old Mennonite family. Until he was sixteen he spent his time at school, on his father's farm, and in his grandfather's distillery, where he kept books. He attended Otterbein University, in Ohio, for a year.

Learning that coke was an essential of the already rapidly developing steel business, Mr. Frick early invested every cent he could in coking coal lands. With the help of an associate of his grandfather, he formed the corporation of Frick & Company, coke dealers, and acquired fifty-one ovens in the Connellsville region and 300 acres of soft-coal lands. During the panic of 1873, with the help of Judge Thomas Mellon, a Pittsburgh banker, he bought out his partners, and while coke was selling at 90 cents a ton enlarged his purchases of suitable lands. Later, the price of coke increased until it was selling at $5 a ton. Before he was thirty years old, Frick was rated a millionaire.

Mr. Frick was a pioneer in modern coke and steel industry and in recent years one of the outstanding financiers of America.

He was a student and lover of art, and by the use of patience and thought and large sums of money he formed one of the finest private collections of paintings, statuary, bronzes, porcelains, enamels, furniture, and other objects of art in existence, all of which, under the provisions of his testament, will in due time be permanently turned over to the public use and enjoyment, together with his costly home in New York, adequately endowed.

Mr. Frick was a Life Member of this Society since its foundation. He died December 2, 1919.

HENRY LAURENCE GANTT

ASSOCIATE

Henry Laurence Gantt was born May 20, 1861, in Calvert County, Md. He was graduated from Johns Hopkins as a Bachelor of Arts when only nineteen years of age and taught for three years at the McDonough School in Baltimore County, which he had attended as a boy. In 1884 he received the degree of Mechanical Engineer from Stevens Institute of Technology.

He was associated with Frederick W. Taylor in his early work at the Midvale and Bethlehem Steel companies, and with this as a basis and his personal ability as an organizer

he later established and successfully conducted his own consulting practice as an industrial engineer. His career was marked by original and thoughtful work, progressive in viewpoint and effect in its accomplishment. In later years he developed a broad conception of industry as a national problem in which he regarded it essential that the man at the top should have the same close scrutiny and careful direction that has in the past been given his co-workers in the lower ranks. To these views he gave expression in a characteristically original manner in many addresses and written articles.

Following his connection with Frederick W. Taylor, he conducted his consulting practice until his death and had as clients more than a score of prominent manufacturing plants. Among these were the Westinghouse Electric and Manufacturing Company, American Locomotive Company, factories of Remington Typewriter Company, Bethlehem Foundry & Machine Company, Saco-Lowell Shops, Ingersoll-Rand Company, Amoskeag Manufacturing Company, Cheney Brothers (silk manufacturers), Brighton Mills, Sayles Bleacheries and Corticelli Silk Mills.

Mr. Gantt was the author of three books: "Work, Wages and Profits: Their Influence on Cost of Living;" "Industrial Leadership" (Yale Lectures); and "Organizing for Work." In addition, he contributed many articles to the technical and daily press.

During the war Mr. Gantt acted in a consulting capacity, first for the Ordnance Department by invitation of General Crozier, who recommended for the entire War Department the use of Gantt's production charts, employed by Mr. Gantt in his work and which had been used in the Ordnance Department. These charts are well known among engineers and constitute an important development in recording the progress of work. When General Crozier was retired as Chief of Ordnance a change in plans was made, but the charts were later used by the U. S. Shipping Board and Emergency Fleet Corporation in routing ships and in following up constructive work.

Mr. Gantt was a Charter Member of this Society. He died November 23, 1919.

FRANK JEFFREY

MEMBER

Frank Jeffrey was born in Aberdeen, Scotland, October 28, 1860. He was apprenticed at fourteen years of age and has since been continuously engaged in shipbuilding in leading Clyde Bank and American shipyards. His work represented all types of deep sea and inland vessels including many well-known commercial and war ships. He came to America in 1881 and was connected with F. W. Wheeler & Company, shipbuilders, Bay City, Mich., from 1885 to 1900, when that plant was taken over by the American

Shipbuilding Company. With the exception of two years as vice-president and general manager of the Union Iron Works, San Francisco, Mr. Jeffrey has been associated with the American Shipbuilding Company as general superintendent at its plants at Bay City, Lorain, and since 1905 of the Detroit Shipbuilding Company, a subsidiary of the American Company. Many ships built by him now in service on the Great Lakes and in all parts of the world are monuments to the memory of one who did things well. Thoroughly honest, capable, and loyal to his friends, associates and his adopted country, he was trusted and beloved by all who knew him. His devotion to his family was exceptional and unvaried. Besides his wife, who was Miss Agnes Corrigan, one daughter, Mrs. Lillie Collins, and one son, William, survive.

Mr. Jeffrey died May 24, 1920.

EDMUND MILLS

MEMBER

The Babcock & Wilcox Company suffered a serious loss of one of its oldest and most faithful employees in the death of Mr. Edmund Mills on January 26, 1920. He was born in Jersey City, N. J., February 15, 1864, so that he was almost fifty-six years of age. Nearly twenty-five years of his life had been spent in the service of The Babcock & Wilcox Company.

He received his education in the public schools of Jersey City, and, having great natural mechanical aptitude, after leaving school served an apprenticeship as a machinist and worked for a short time at that trade. Later he was in business for himself for nearly four years. He was then for about two years in the draughting department of the Daft Electric Company, about a year of the time as chief draughtsman. From 1891 to 1895 he was with the Colwell Iron Works, which later became the Wheeler Condenser and Engineering Company.

In 1895 Mr. Mills entered the marine department of The Babcock & Wilcox Company as a draughtsman. He assisted Mr. W. D. Hoxie in the development of the Babcock & Wilcox marine boiler as it now exists by working out the detail drawings to embody Mr. Hoxie's ideas. He was later made chief draughtsman of the marine department. In 1909 he was transferred to the general office of the company as an engineer and correspondent, and here the last eleven years of his life were spent in most efficient work.

He not only had fine mechanical judgment, but he had equipped himself thoroughly as an engineer dealing with combustion and the generation of steam, so that he was well equipped for the work which devolved upon him. He was conscientious and careful

in the highest degree so that it was always safe to feel that, when he had completed the work on any problem, it was well done. His intimate service during almost the entire life of the marine department had given him a familiarity with all the marine work done by the company which those coming later on can never hope to equal. His memory was remarkable, and until the enormous output during the World War it was safe to say that he knew the number of every marine job and a good deal of its history during the life of the vessel.

Besides his life work as an engineer Mr. Mills had interests for his leisure hours outside of business in which the same painstaking attention to detail gave him rather unusual distinction. His chief avocation was the study of polar exploration, particularly the Arctic. He had been collecting books on this subject for years, and his collection is probably the finest and most complete of any private individual. He enjoyed the acquaintance of the late Admiral Melville, and on one occasion showed the admiral the catalogue of his Arctic library. After looking it over the admiral told him that included in it were a number of books of which he had heard but which he had never seen himself. It had been his intention to present the collection to one of the geographic societies or to some public library, and his family has carried out this intention.

He was also interested in astronomy as an amateur and had built a small observatory at his home, where, with a telescope of moderate aperture, he carried on searches for double stars and other astronomical pursuits in which an amateur with a small equipment can be of service.

Mr. Mills was a fine specimen of a Christian gentleman of the highest type and was a faithful friend and an agreeable companion to all his associates. He had been an officer and active worker for many years in one of the leading churches of Jersey City and always stood for what was finest and best in American manhood. He leaves behind him an enviable record as a loyal, efficient worker and as an upright, agreeable man whose memory will always be cherished by those who were close to him as one of the agreeable features of their lives.

WILLIAM T. NEVINS

MEMBER

William T. Nevins was born at Cleveland, Ohio, January 6, 1873. He was educated in the schools of Cleveland and served an apprenticeship at the trade of shipfitter at the Globe Iron Works, Cleveland, from 1888 to 1891. Thereafter he was employed as journeyman in various coast and lake yards and navy yards from 1891 to 1896. He served as foreman of hull construction and repair department of the Globe Iron Works,

Cleveland, and the Cleveland Shipbuilding Company, Lorain, Ohio, from 1896 to 1899. He was assistant to Robert Logan, naval architect, from 1899 to 1901; assistant superintendent of the Buffalo Dry Dock Company from 1901 to 1902, and superintendent of that company from 1902 to 1904.

Following the above extensive period of practical experience, he engaged in the business of naval architect and surveyor from 1904 to 1918 at Buffalo, Chicago, San Francisco and Seattle. From 1918 to 1920 he was employed as superintendent of ship construction, Chickasaw Shipbuilding Company, and at the time of his death as hull constructor, Pensacola Shipbuilding Company, Pensacola, Fla.

Mr. Nevins died April 29, 1920.

GUSTAV ZENO PROCHAZKA

MEMBER

Gustav Zeno Prochazka was born February 12, 1875, at Dolni, Bobrova, Moravia. He graduated from the high school in Prague in 1892 and the School of Technology, M.E., in 1897. From 1898 to 1903 he was a naval constructor in the Austro-Hungarian Navy and thereafter from 1904 to 1910 he held various responsible technical positions in his native country. In 1915 he came to this country, and until 1915 he held the position of assistant engineer with the Hon. Gustav Lindenthal. From 1915 to 1918 he was employed as hull draughtsman at the Fore River Plant, Bethlehem Shipbuilding Company, and from 1918 until his death, November 19, 1919, he was employed as chief hull draughtsman by the Foundation Company at New Orleans, La.

ALVIN A. WINSHIP

MEMBER

Alvin A. Winship was born September 8, 1873, at Brooklyn, N. Y., and was educated in the schools of Brooklyn and San Francisco.

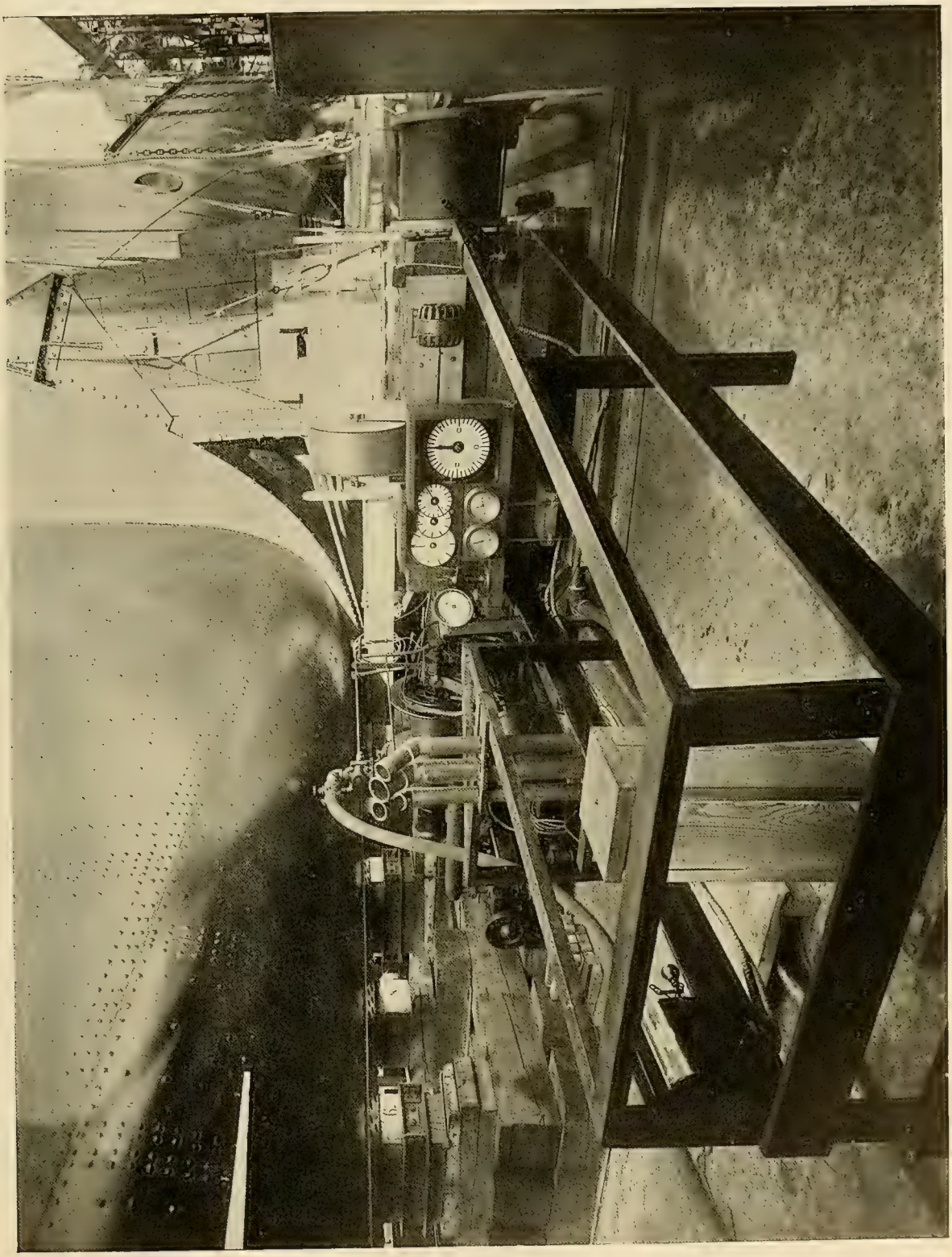
He served an apprenticeship of four years, from 1889 to 1893, at Davidson Pump Works, Brooklyn, N. Y., and was thereafter employed on naval trial trips for four years; foreman of the ship machinists' department in the shipyard at Sparrows Point for eighteen years. He had a chief engineer's license for ocean steamships for sixteen years and for four years was superintendent of machinery at the Bethlehem Shipbuilding Corporation, Sparrows Point, Md., which position he held at the time of his death.

Mr. Winship died October 2, 1920.

To illustrate paper on "Launching of Ships in Restricted Waters," by Lieutenant-Commander H. E. Saunders, Construction Corps, U. S. N., Member.

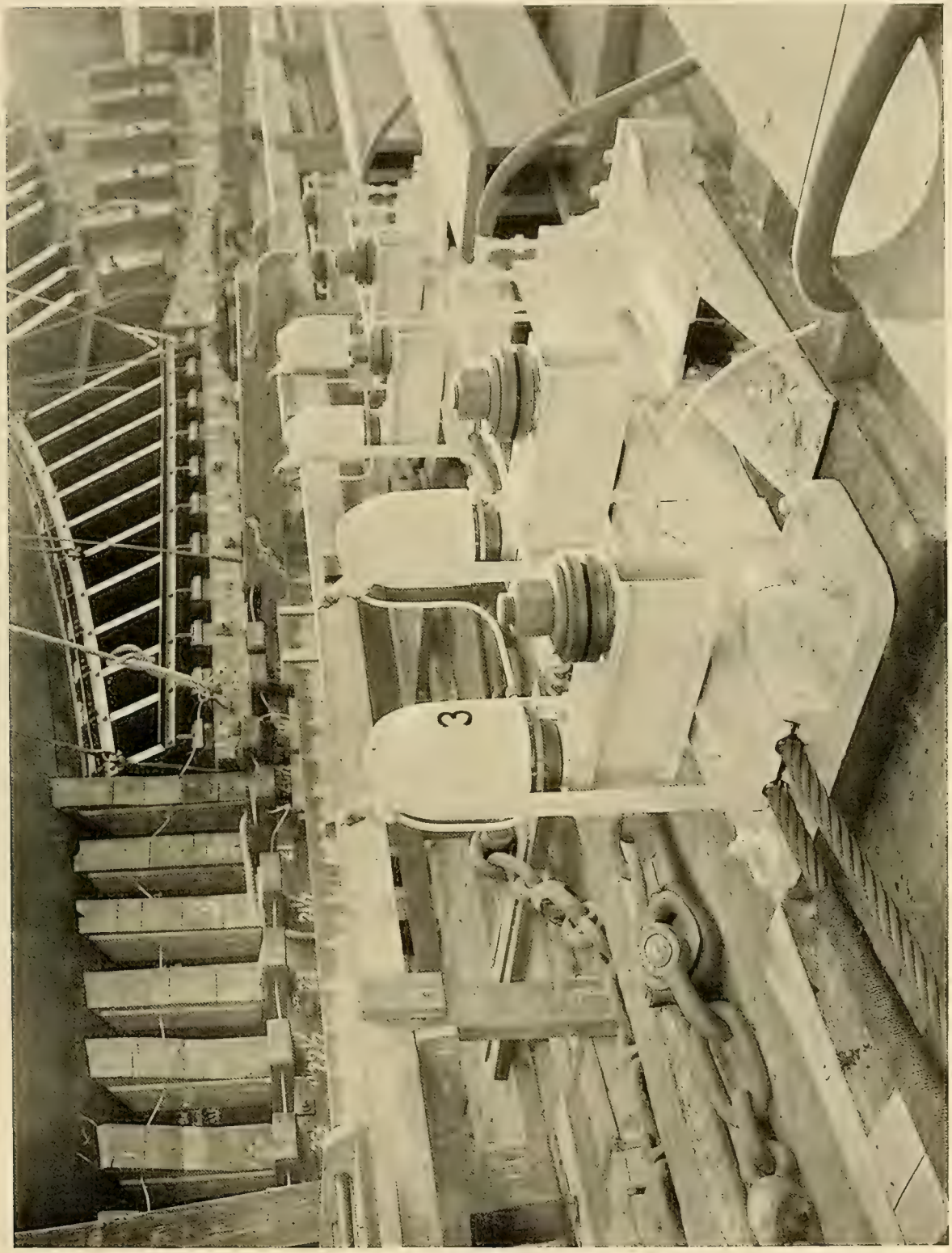
U. S. S. California. Launching, Navy Yard, Mare Island, Cal., November 20, 1919.

To illustrate paper on "Launching of Ships in Restricted Waters," by Lieutenant-Commander H. E. Saunders, Construction Corps, U. S. N., Member.



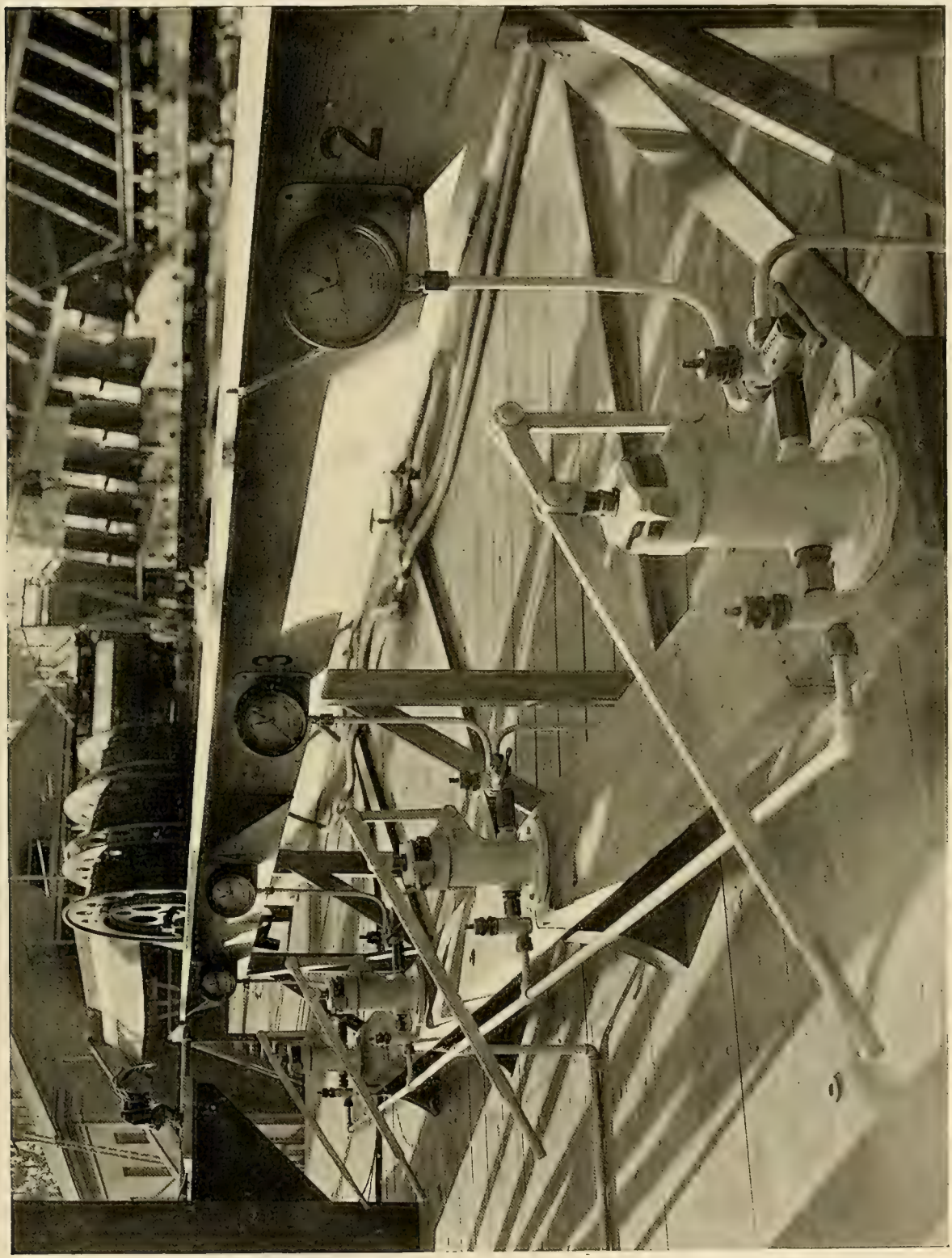
Brake Control Station.

To illustrate paper on "Launching of Ships in Restricted Waters," by Lieutenant-Commander H. E. Saunders, Construction Corps, U. S. N., Member.



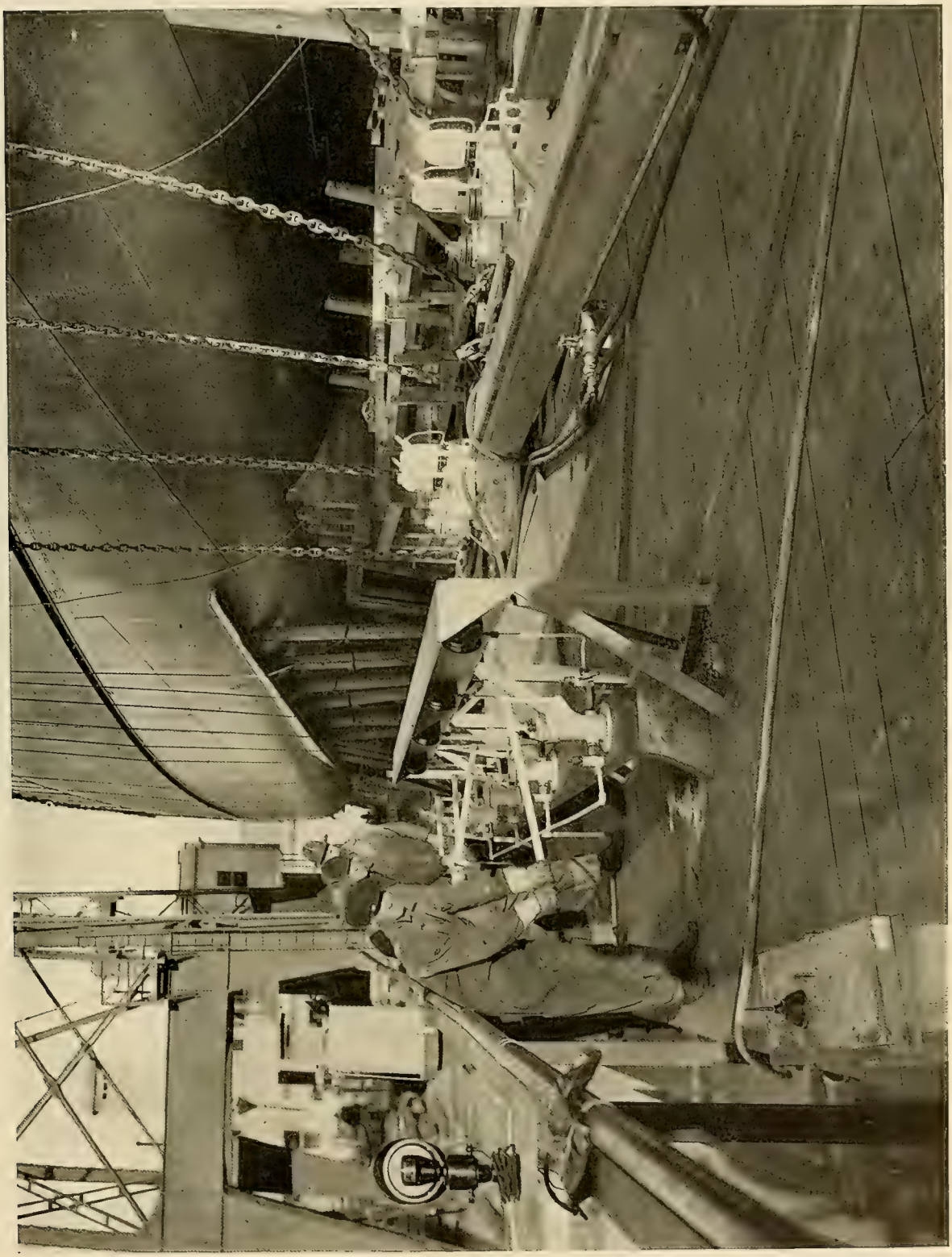
Number 3 Brake, Starboard Side.

To illustrate paper on "Launching of Ships in Restricted Waters," by Lieutenant-Commander H. E. Saunders, Construction Corps, U. S. N., Member.



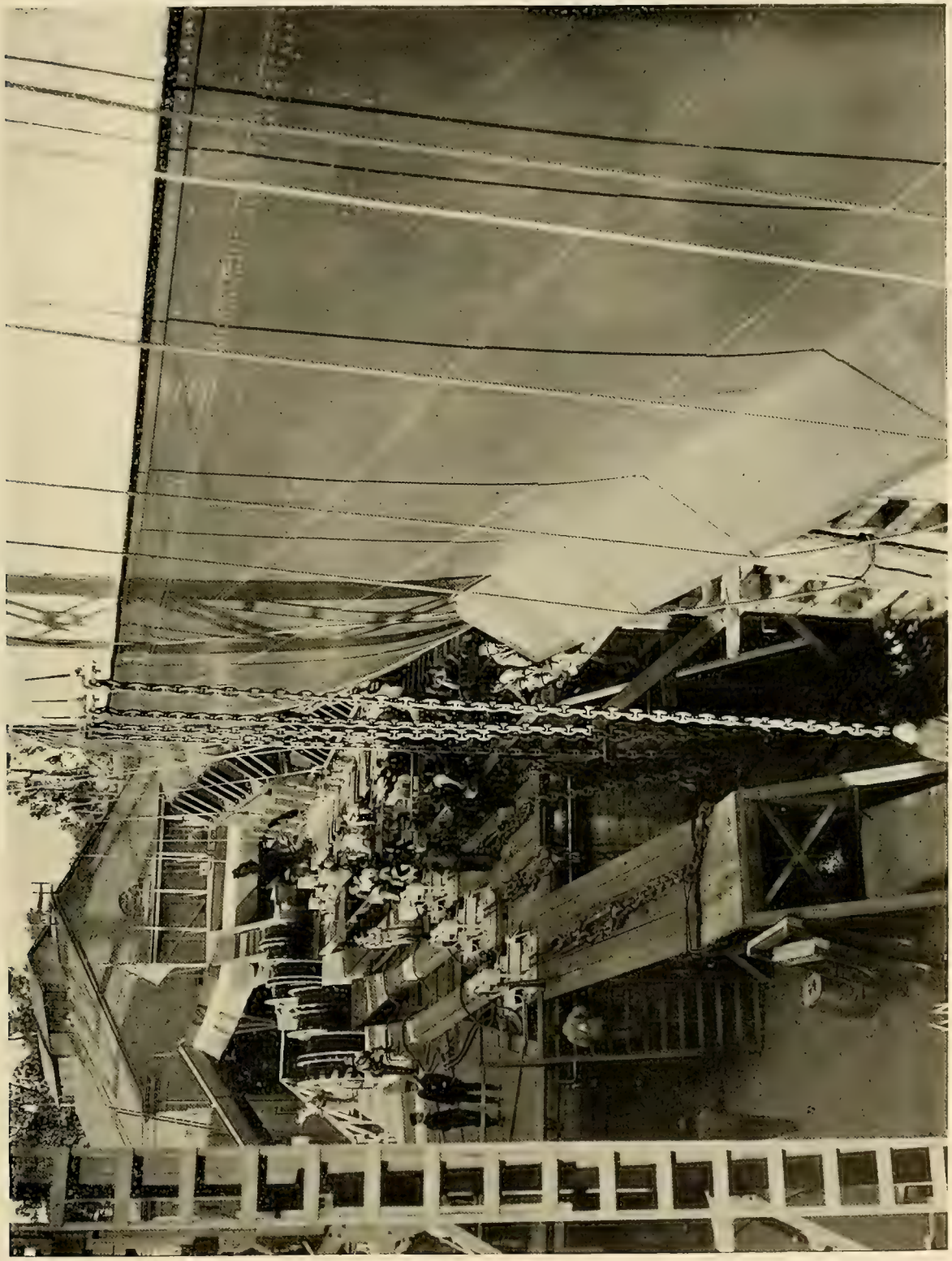
Hand Pump and Gauges, Port Side.

To illustrate paper on "Launching of Ships in Restricted Waters," by Lieutenant-Commander H. E. Saunders, Construction Corps, U. S. N., Member.



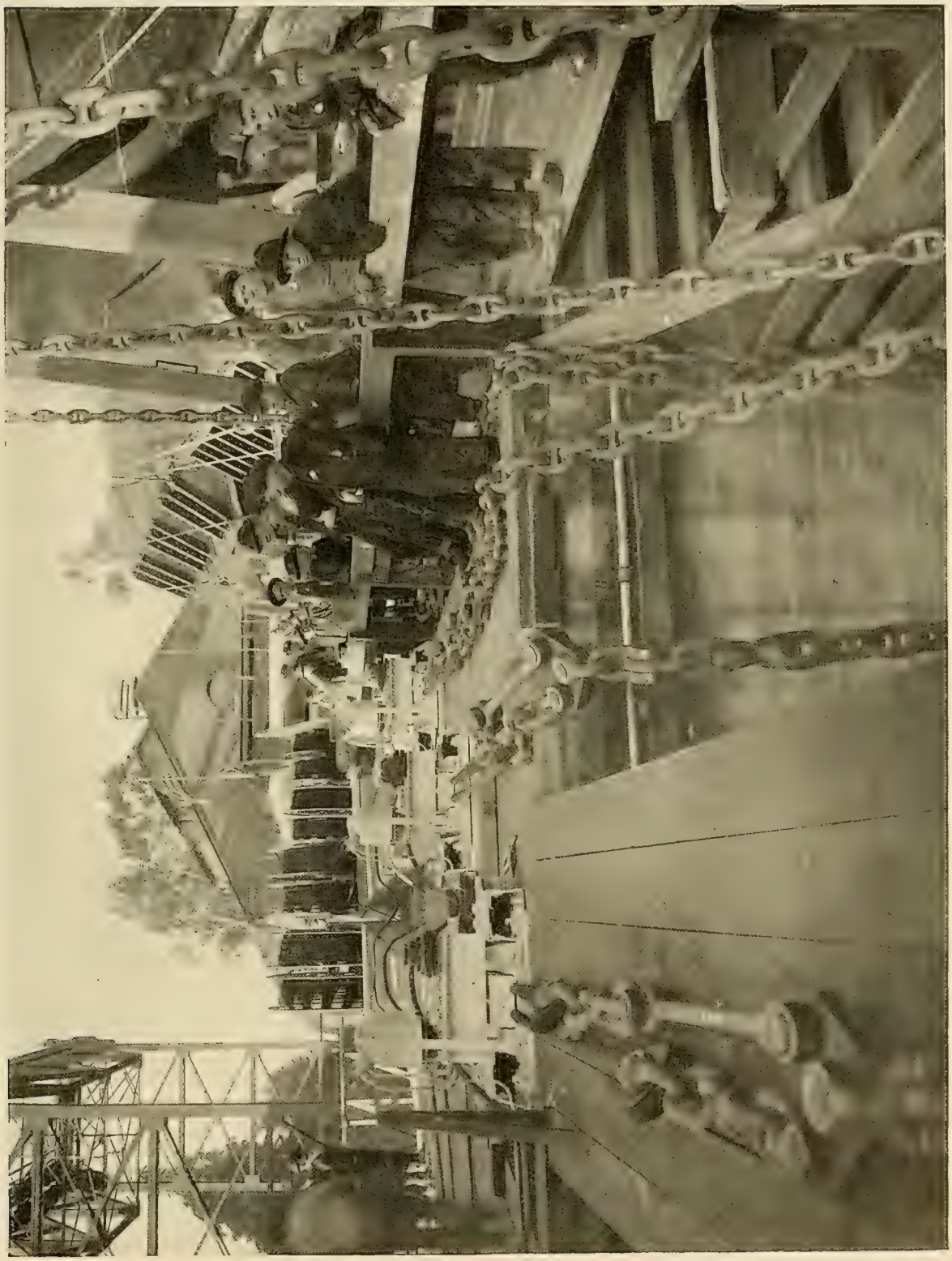
Brake Crews, Starboard Side.

To illustrate paper on "Launching of Ships in Restricted Waters," by Lieutenant-Commander H. E. Saunders, Construction Corps, U. S. N., Member.



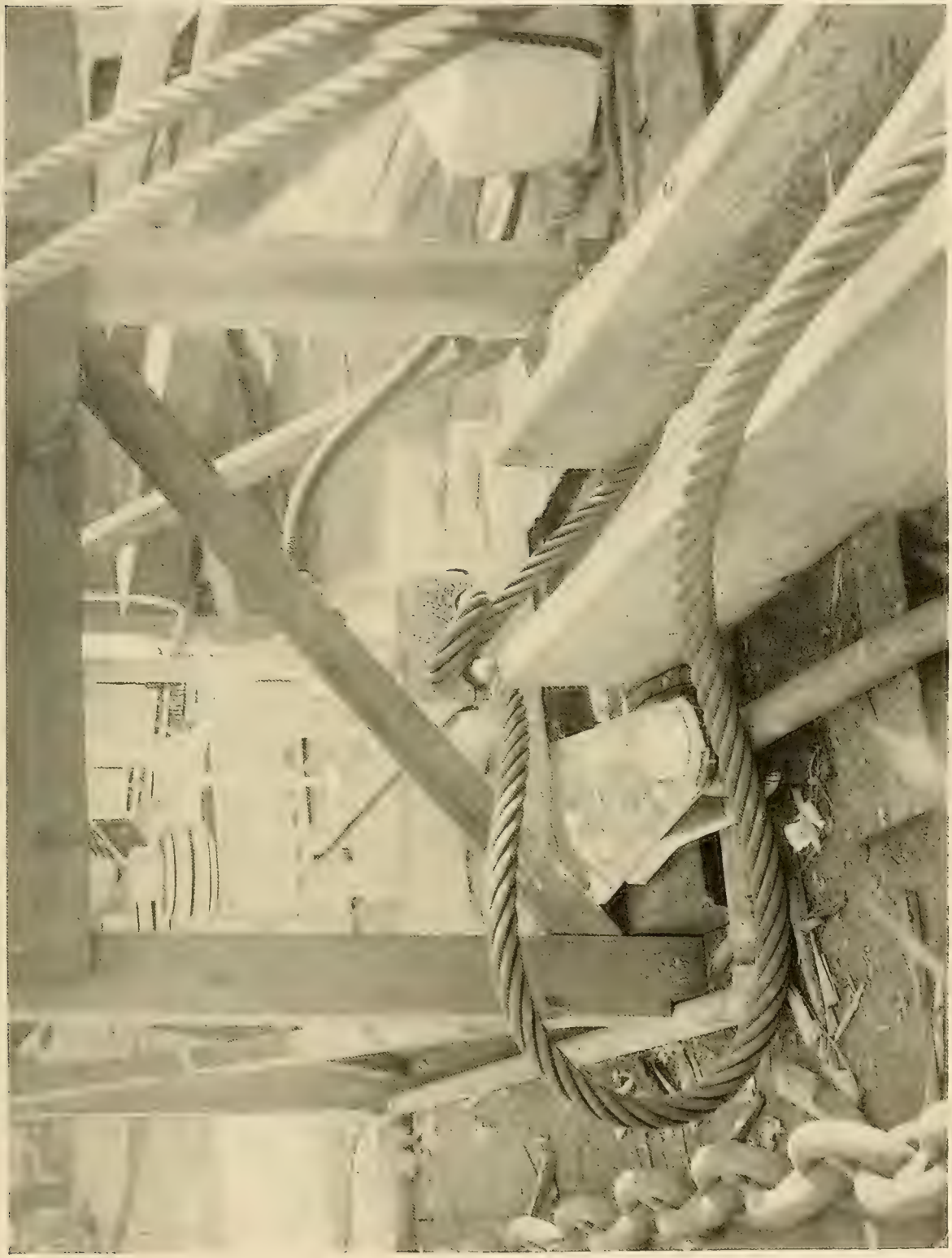
General View, Port Side.

To illustrate paper on "Launching of Ships in Restricted Waters," by Lieutenant-Commander H. E. Saunders, Construction Corps, U. S. N., Member.



Brakes and Chains, Port Side.

To illustrate paper on "Launching of Ships in Restricted Waters," by Lieutenant-Commander H. E. Saunders, Construction Corps, U. S. N., Member.

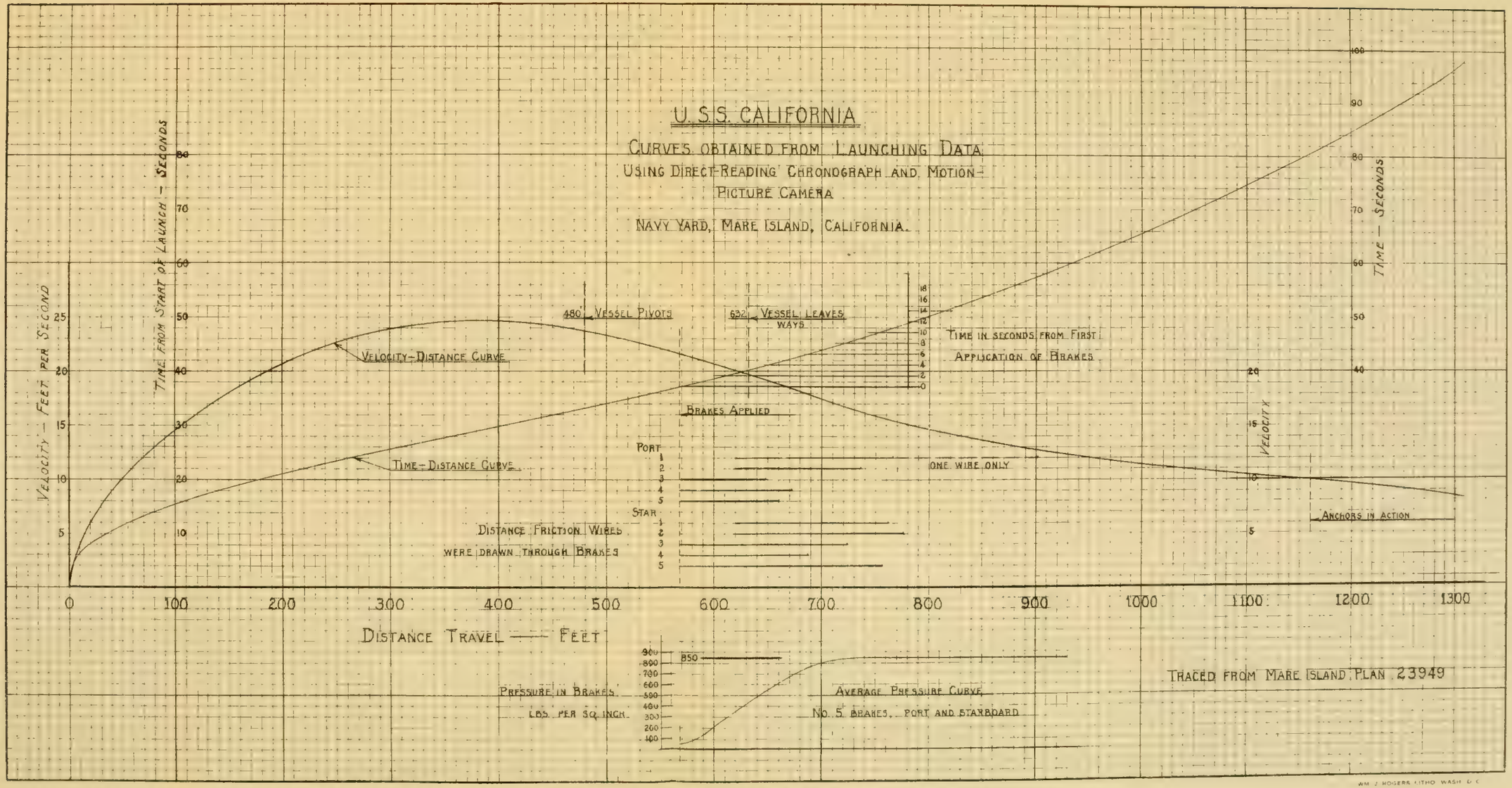


Number 1 Starboard Brake After Launch.

To illustrate paper on "Launching of Ships in Restricted Waters," by Lieutenant-Commander H. E. Saunders, Construction Corps, U. S. N., Member.

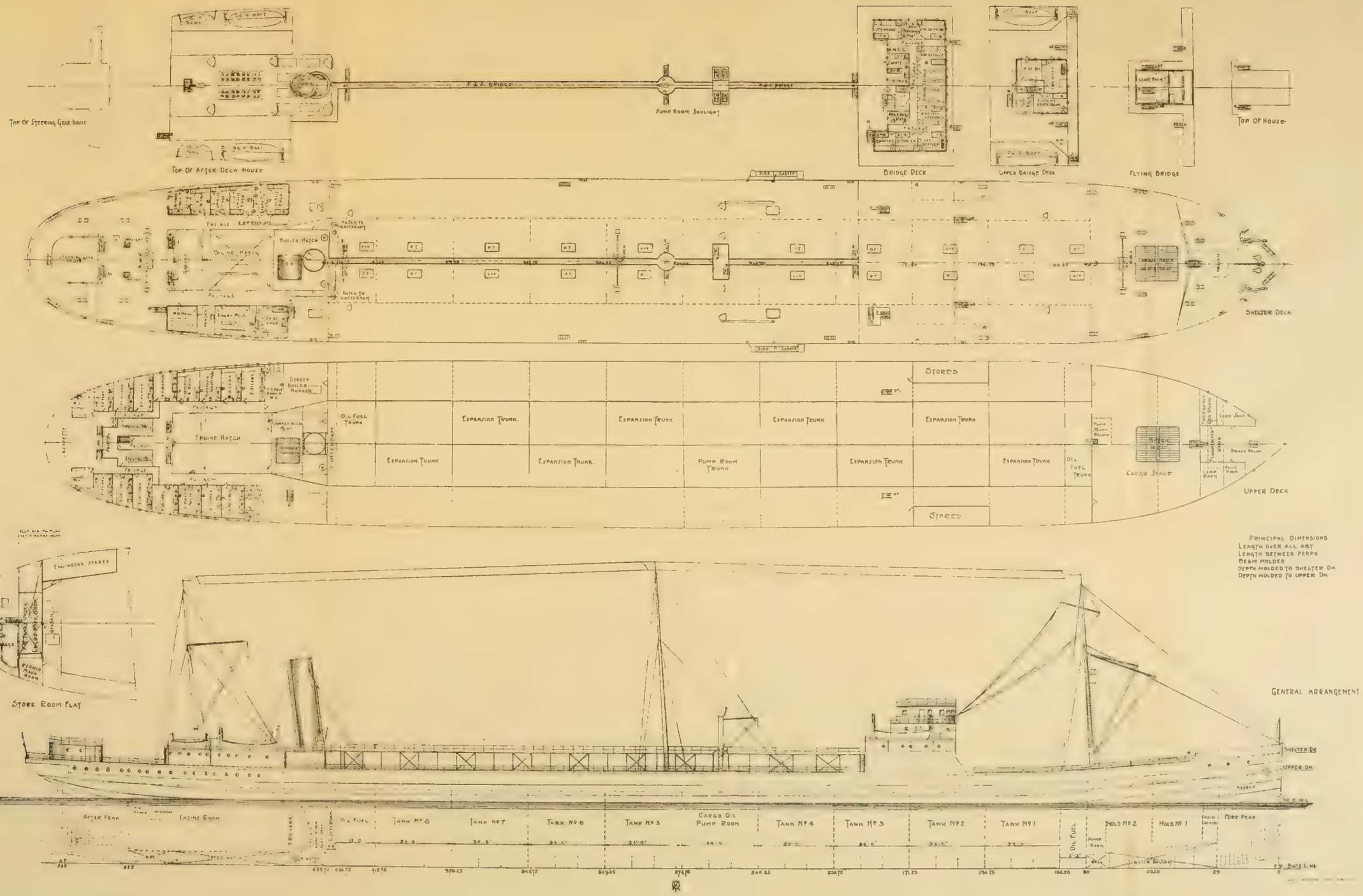
BROKEN LINKS IN CHAINS OF LAUNCHING BRAKES.

To illustrate paper on "Launching of Ships in Restricted Waters,"
by Lieutenant-Commander H. E. Saunders, Construction Corps, U. S. N., Member.



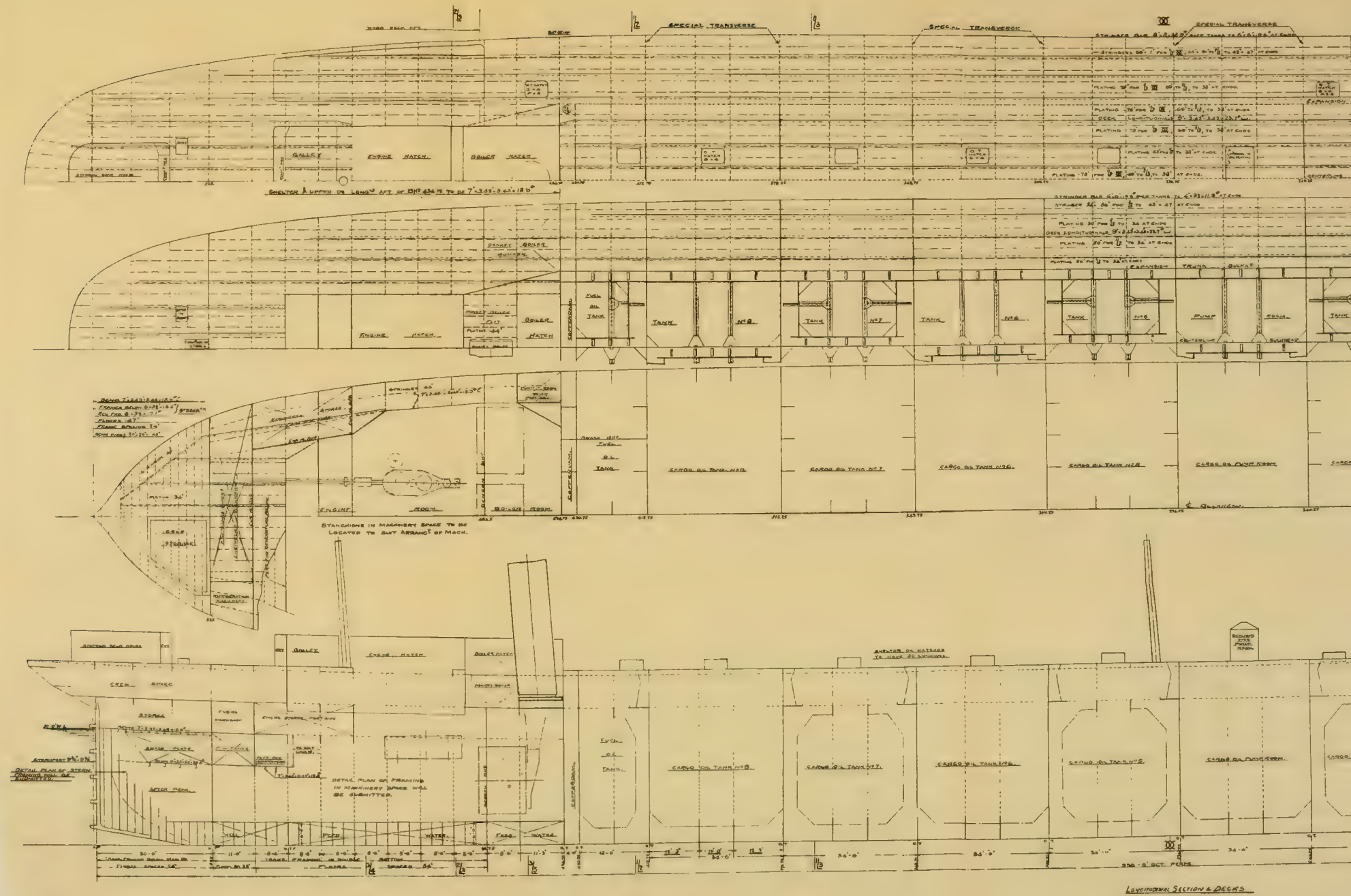
U.S.S. CALIFORNIA - Curves obtained from launching data.

To illustrate paper on "New 20,000-Ton Tankers," by H. F. Norton, Esq., Member.



Principal Dimensions	
Length over all abt	572 - 0
Length between perps	555 - 0
Beam molded	75 -
Depth molded to shelter Dk	43 - 3
Depth molded to upper Dk	33 -

To illustrate paper on "New 20,000-Ton Tankers," by H. F. Norton, Esq., Member.



LONGITUDINAL SECTION & DECKS

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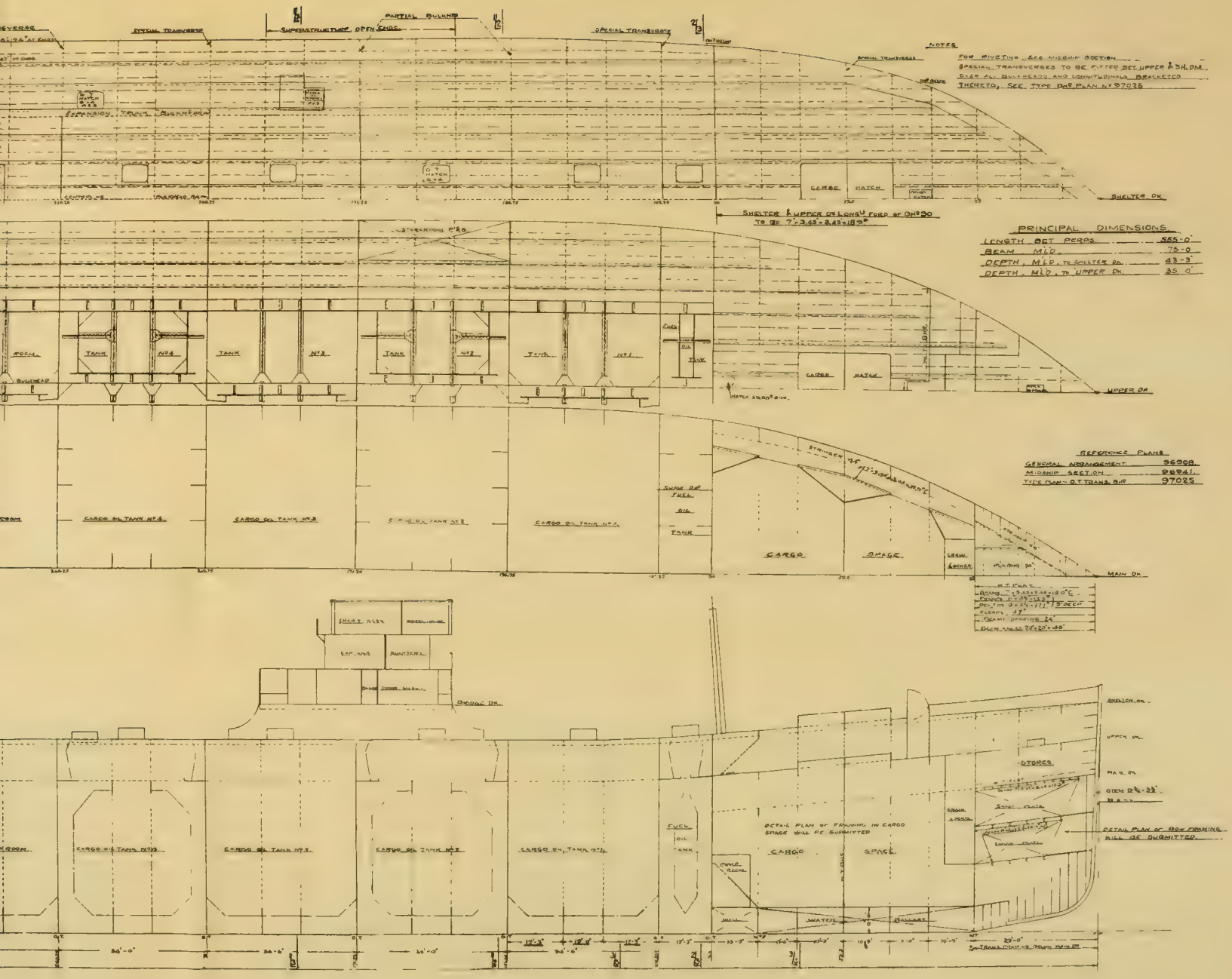
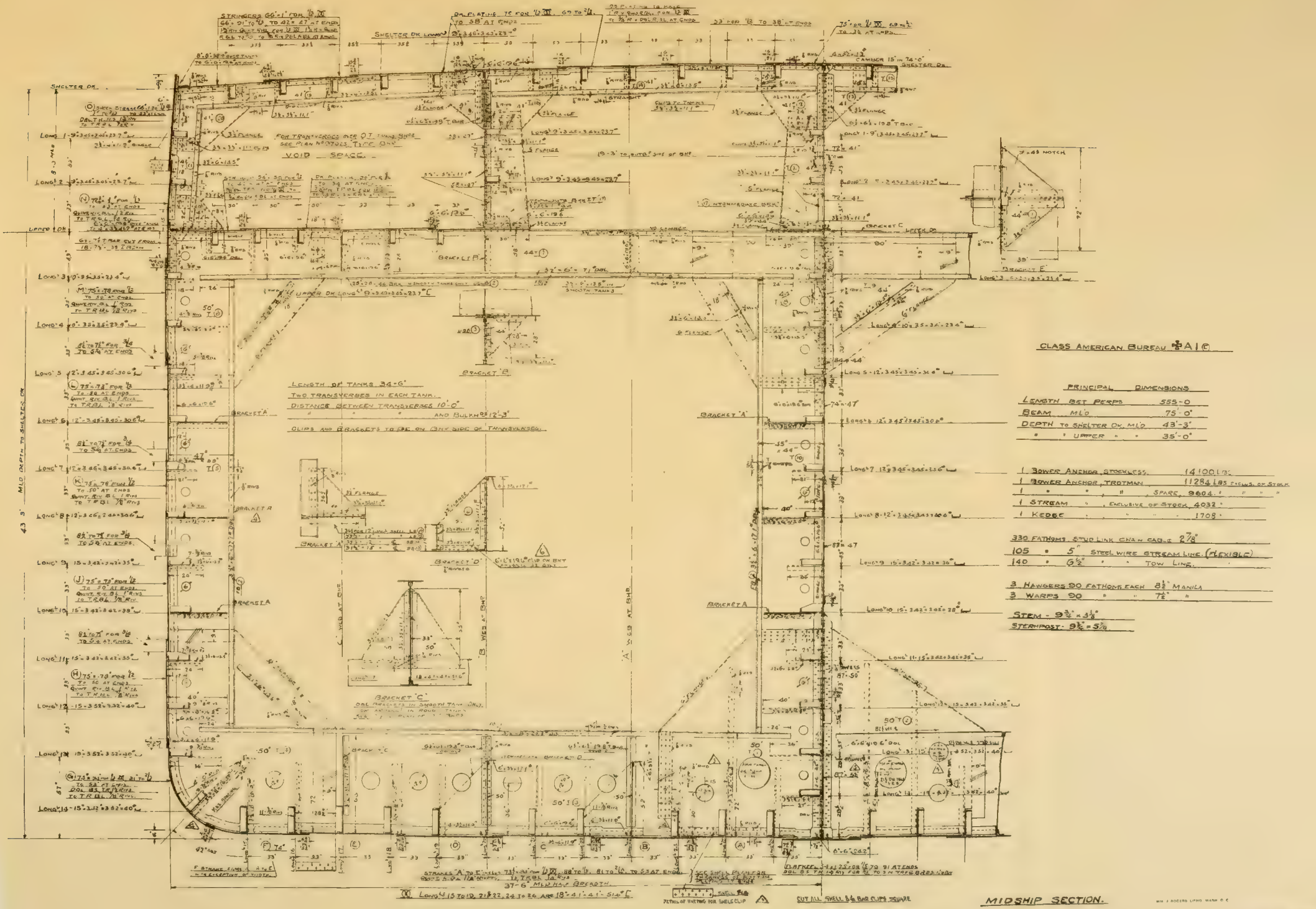
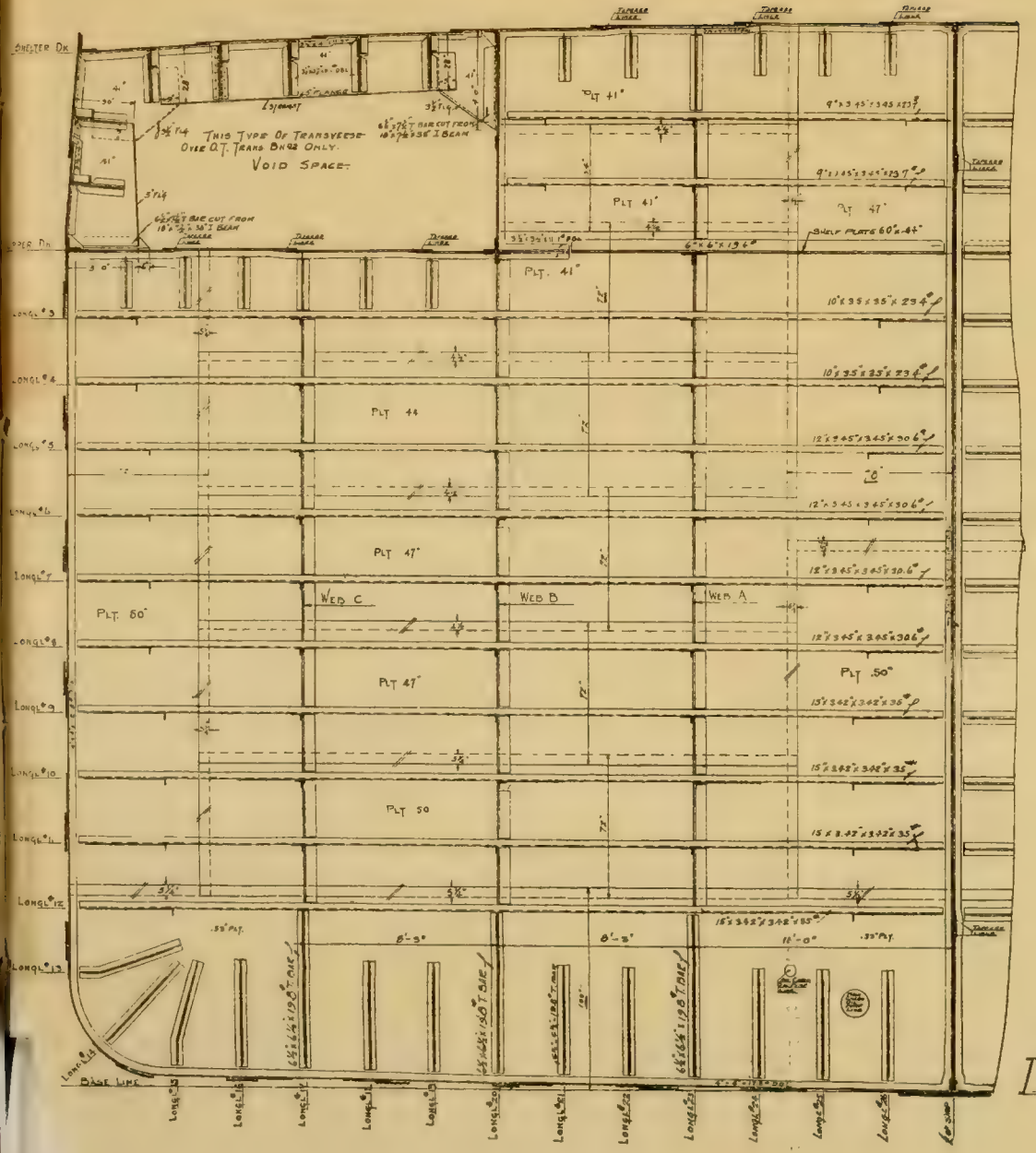


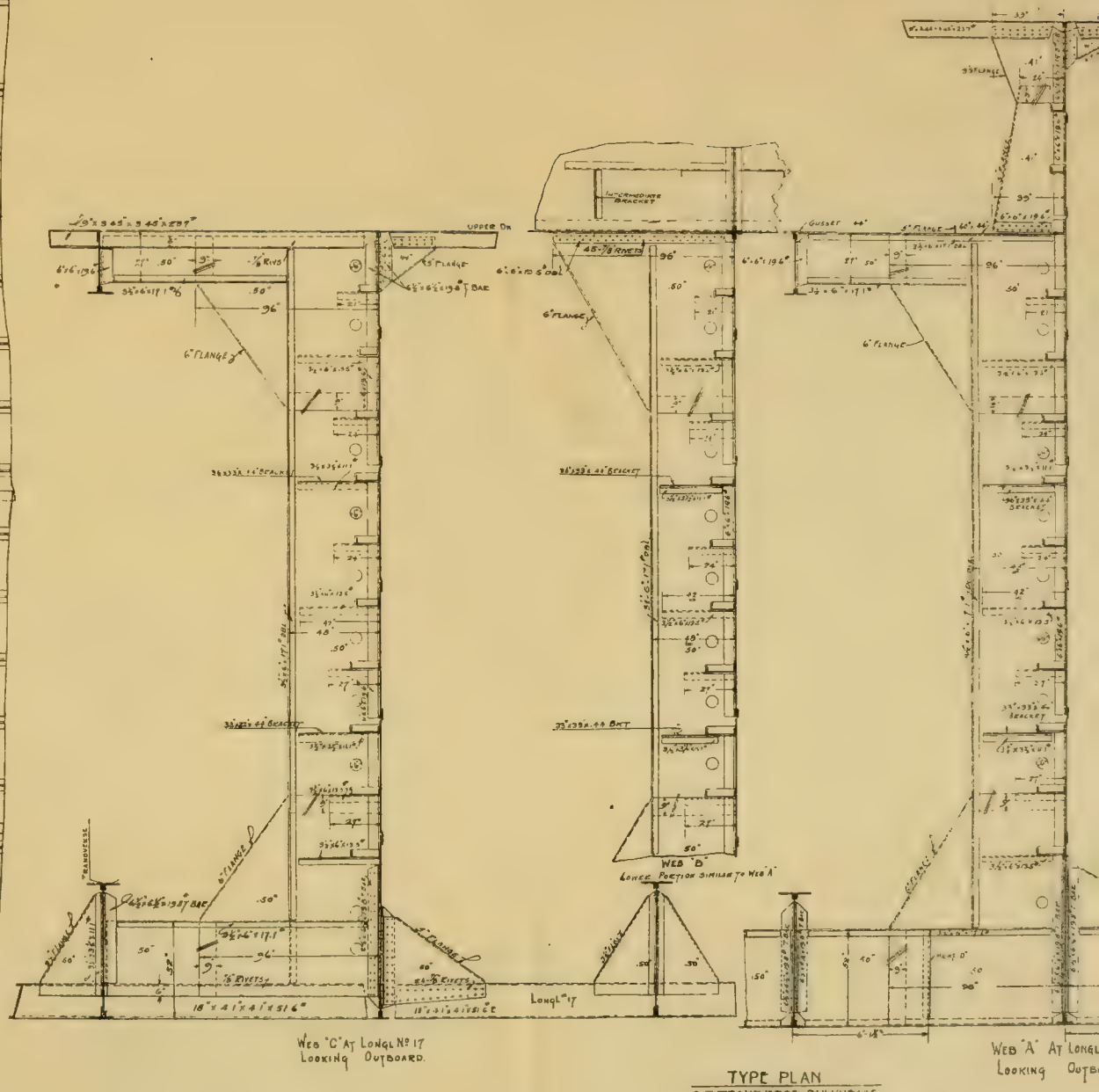
WM J ROGERS, LITHO WASH D C

To illustrate paper on "New 20,000-Ton Tankers,"
by H. F. Norton, Esq., Member.



To illustrate paper on "New 20,000-Ton Tankers," by H. F. Norton, Esq., Member.

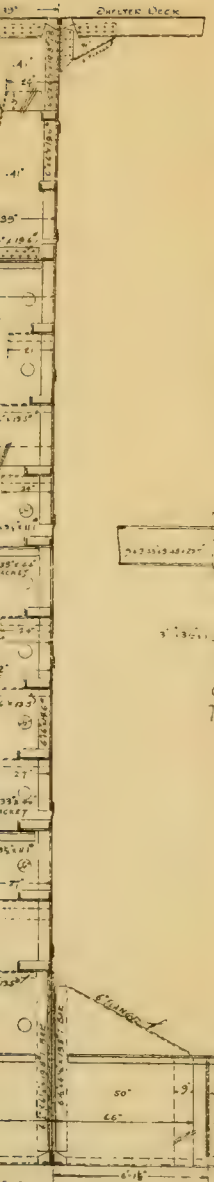


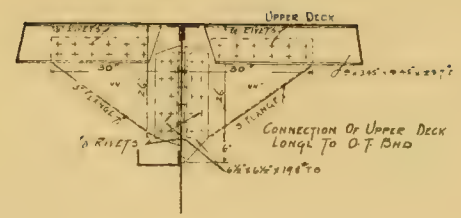


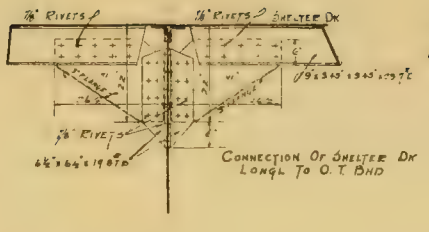
WEB "C" AT LONGL Nº 17
LOOKING OUTBOARD.

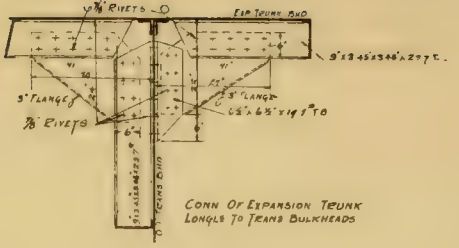
WEB "A" AT LONGL
LOOKING OUTB

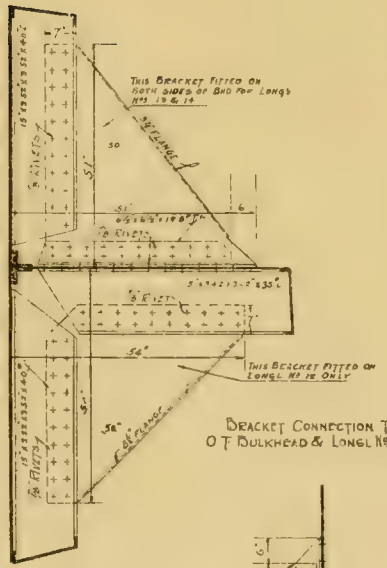
TYPE PLAN
O·T·TRANSVERSE BULKHEADS

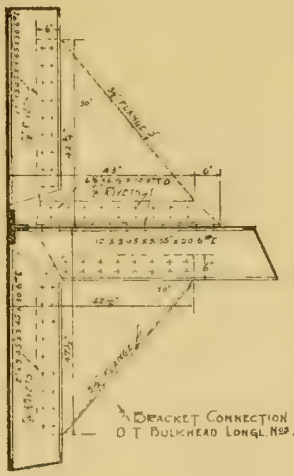


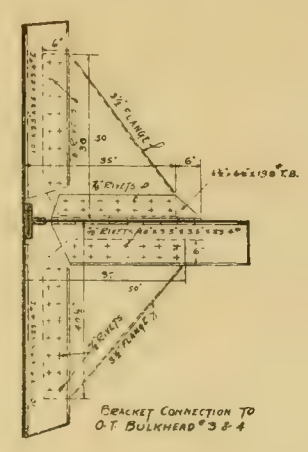


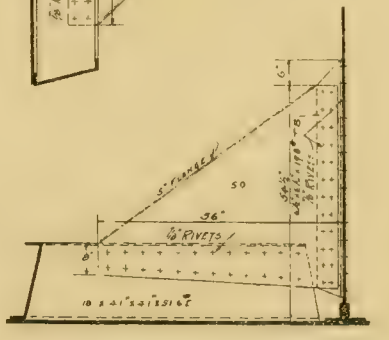


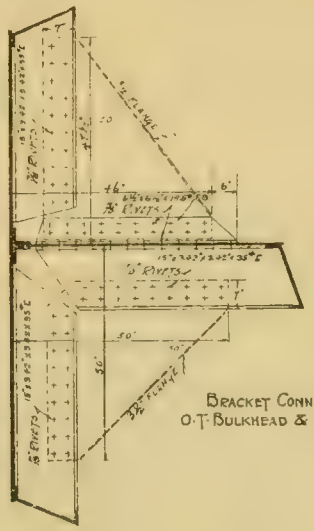






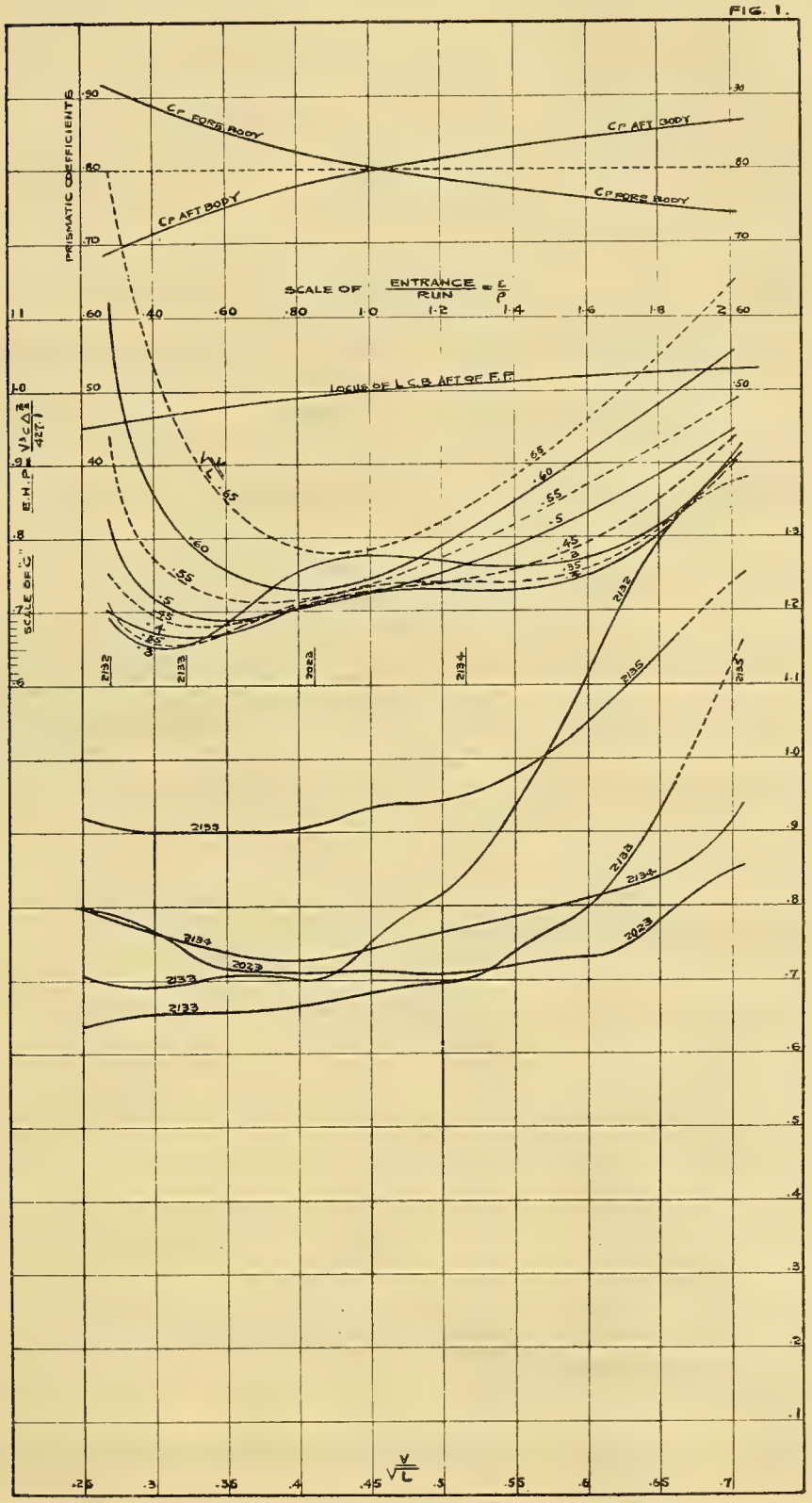






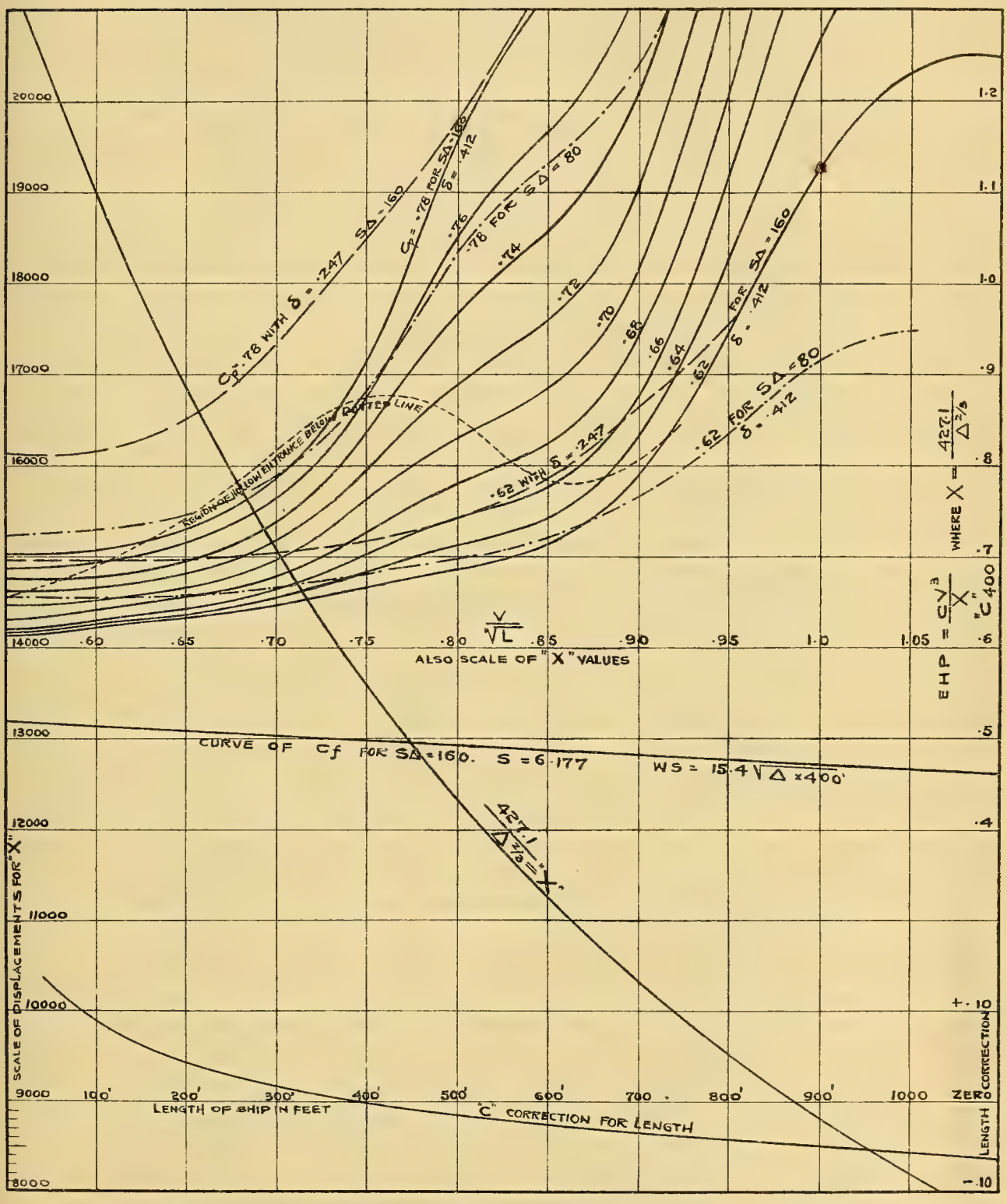
Wm J Rogers Litho Wash D C

To illustrate paper on "Economical Cargo Ships—Some Model Experiments," by Alfred J. C. Robertson, Esq., Member.



To illustrate paper on "Economical Cargo Ships—Some Model Experiments," by Alfred J. C. Robertson, Esq., Member.

FIG. 2.



WM J. ROGERS, LITHO WASH., D. C

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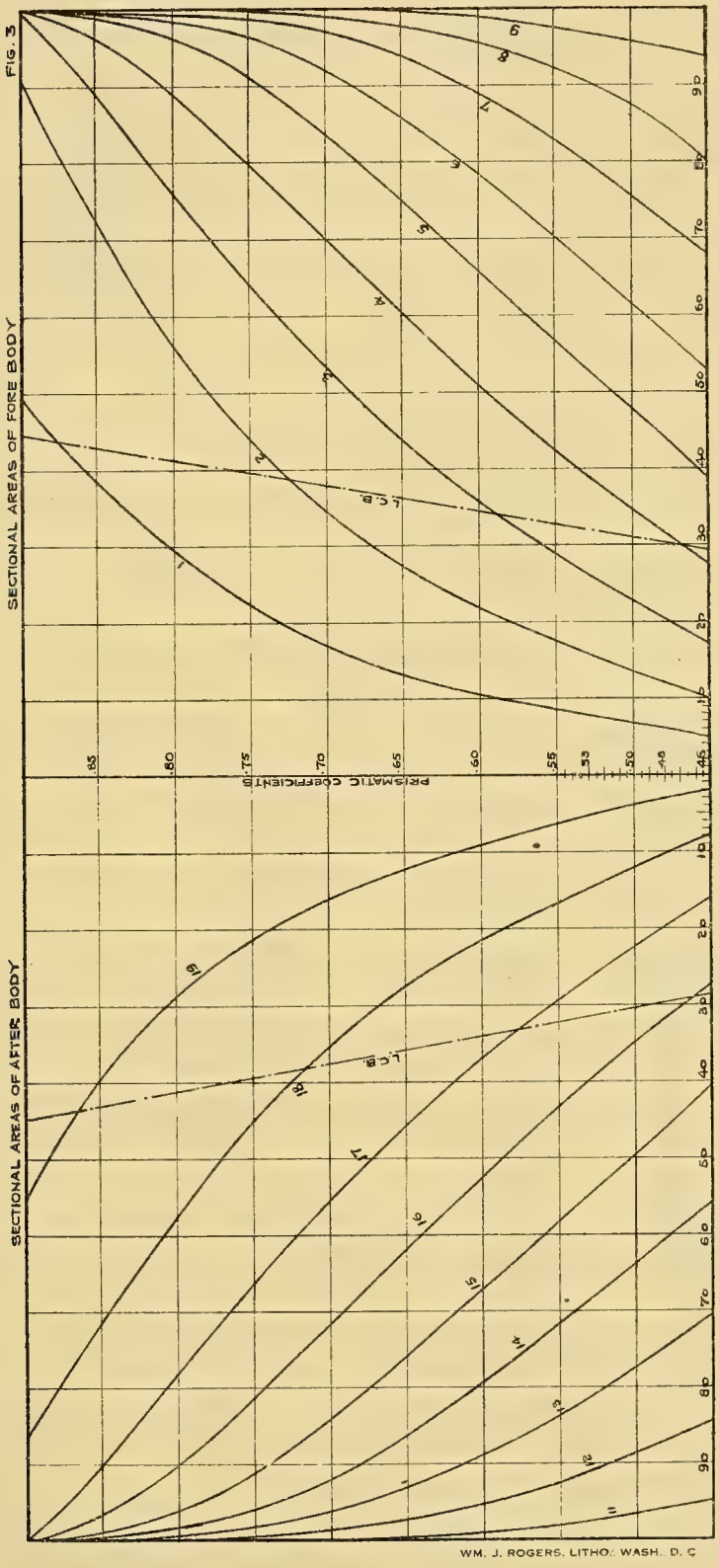

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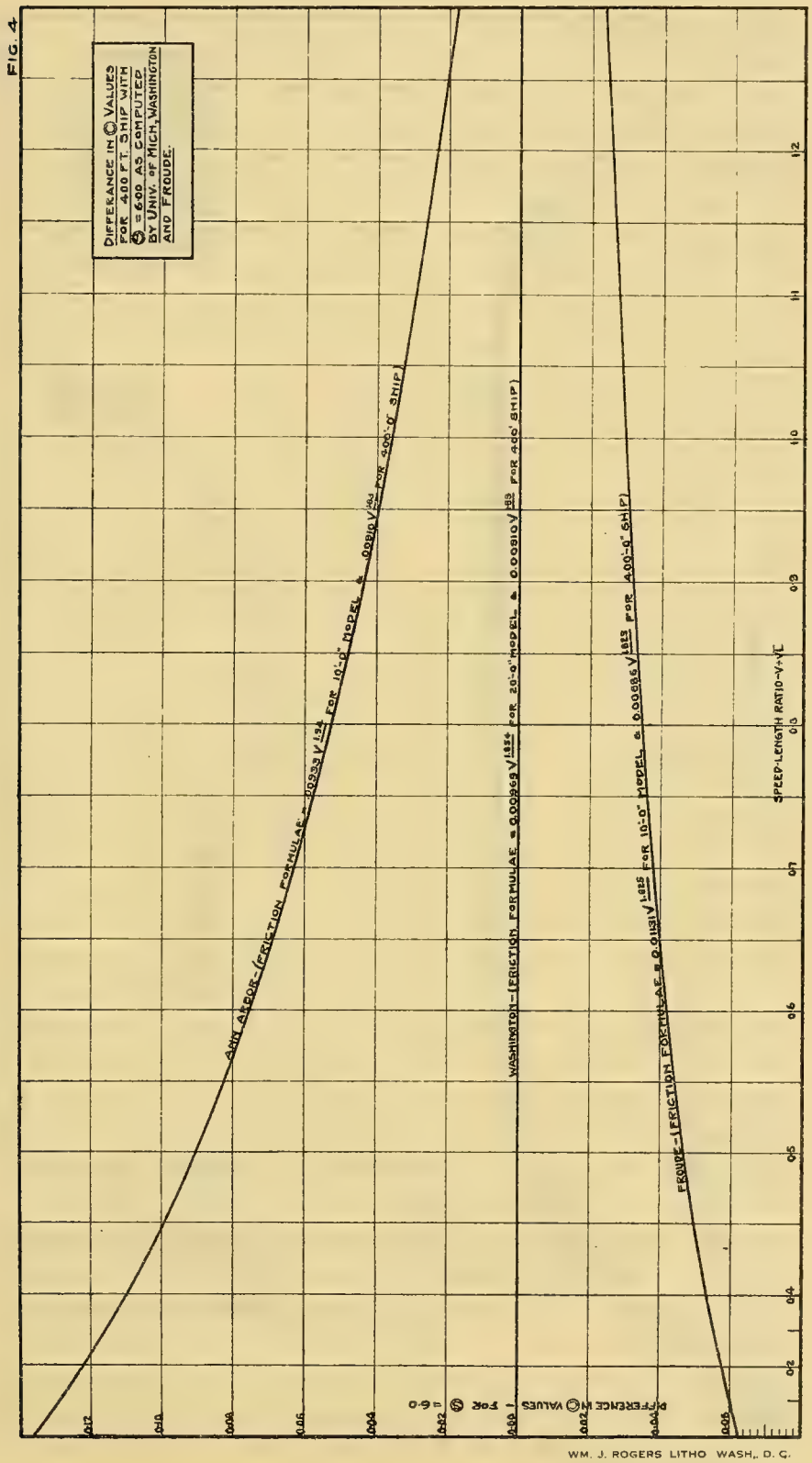

WM. J. ROGERS LITHO WASH., D. C.

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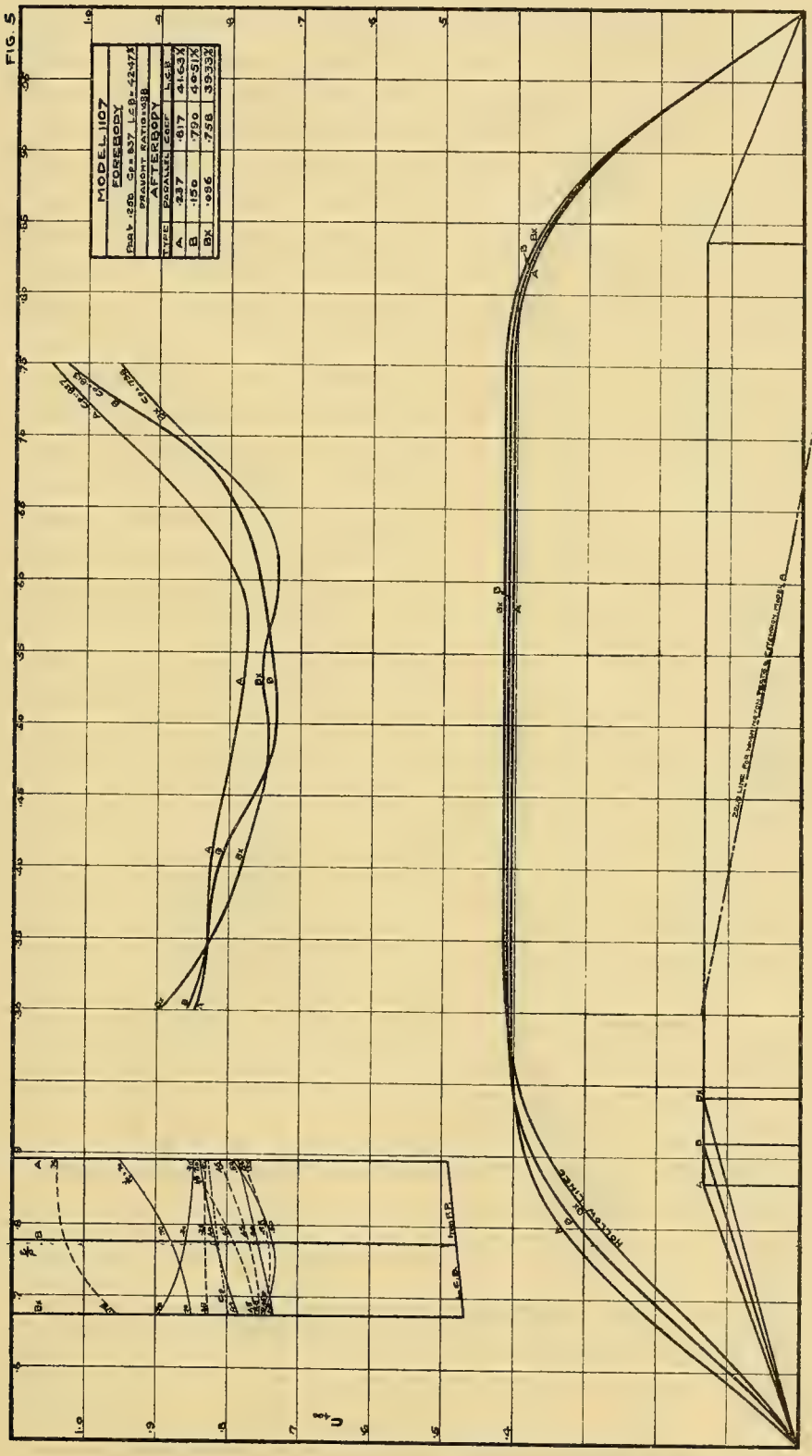

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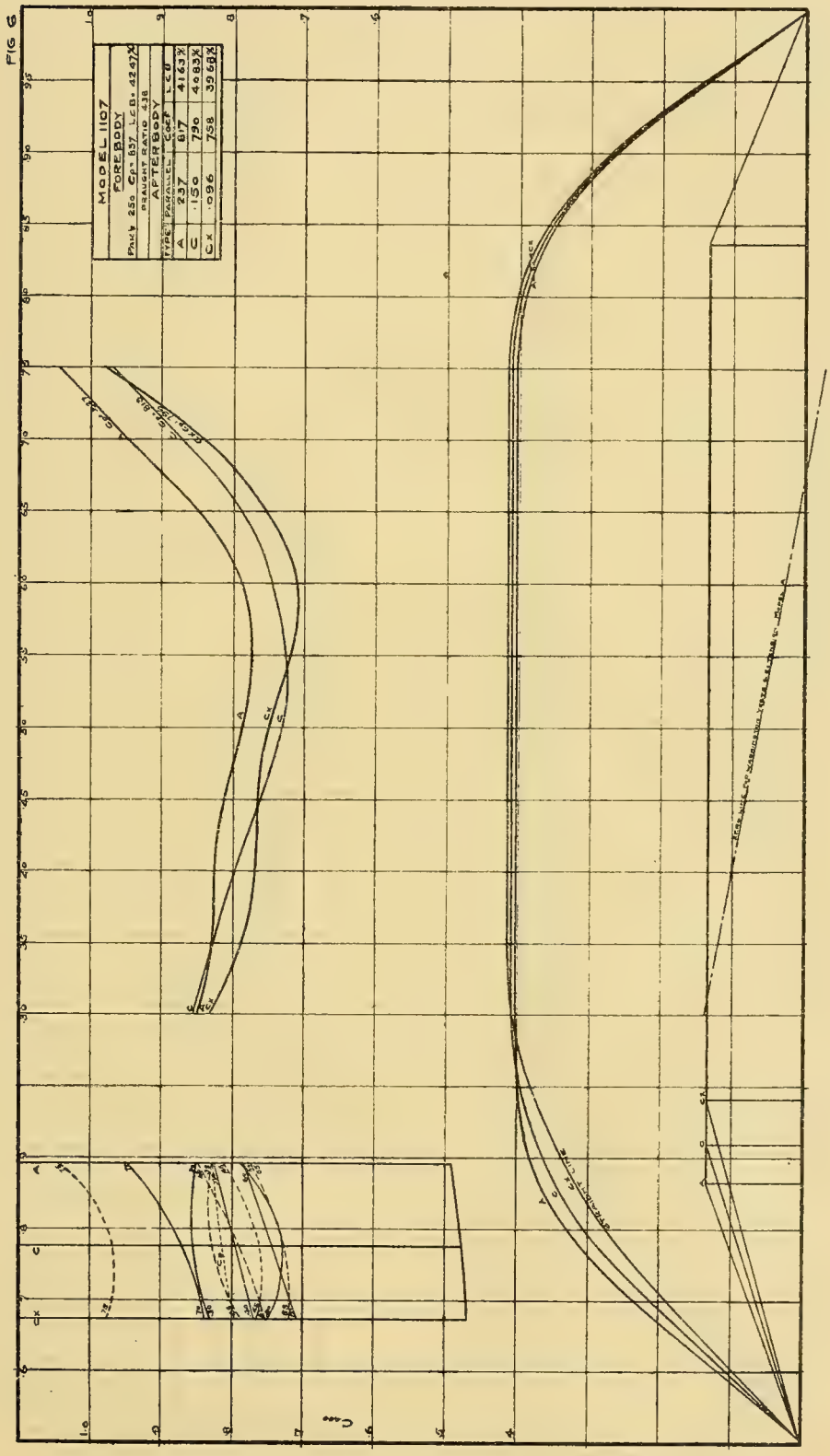

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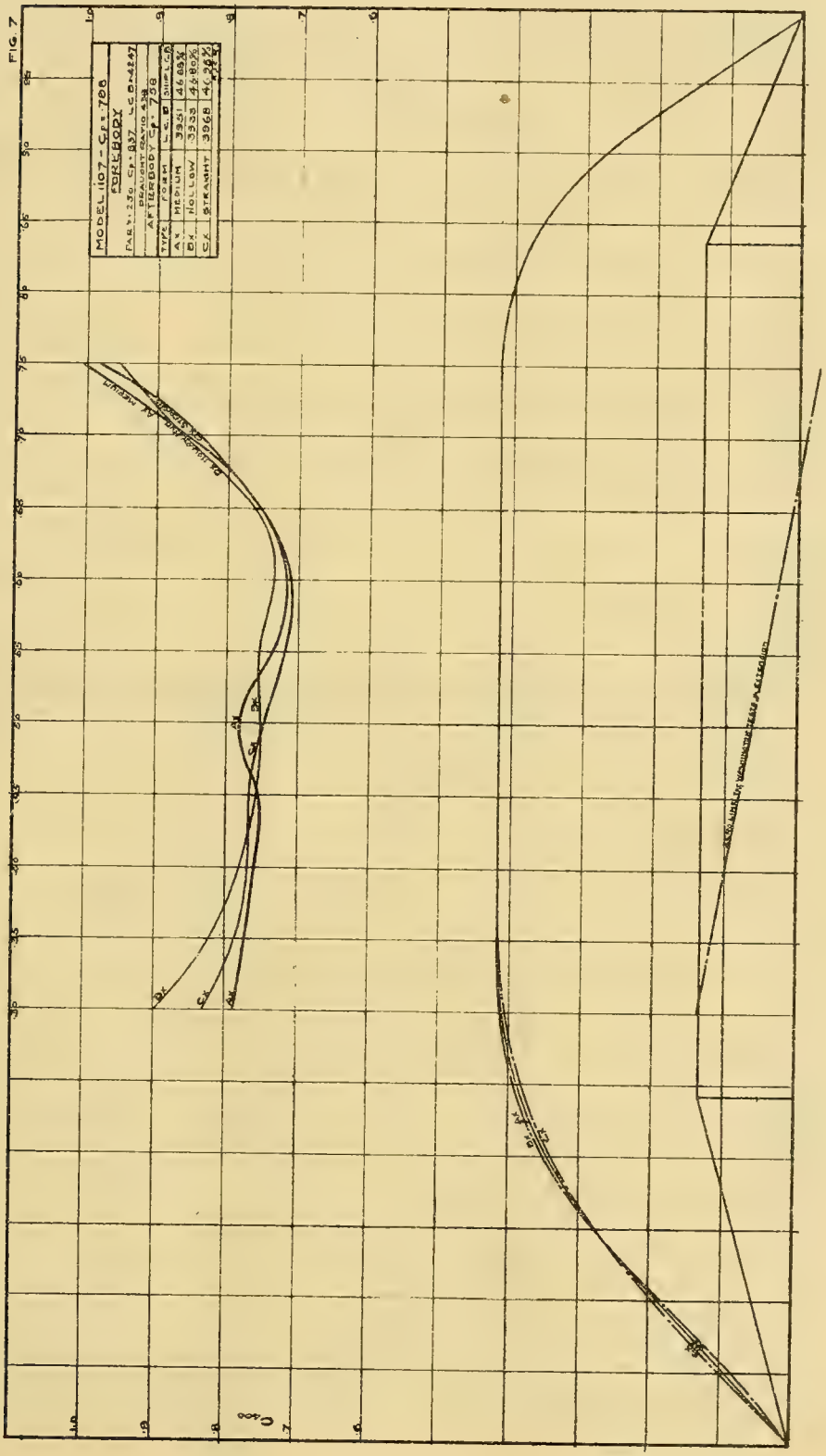
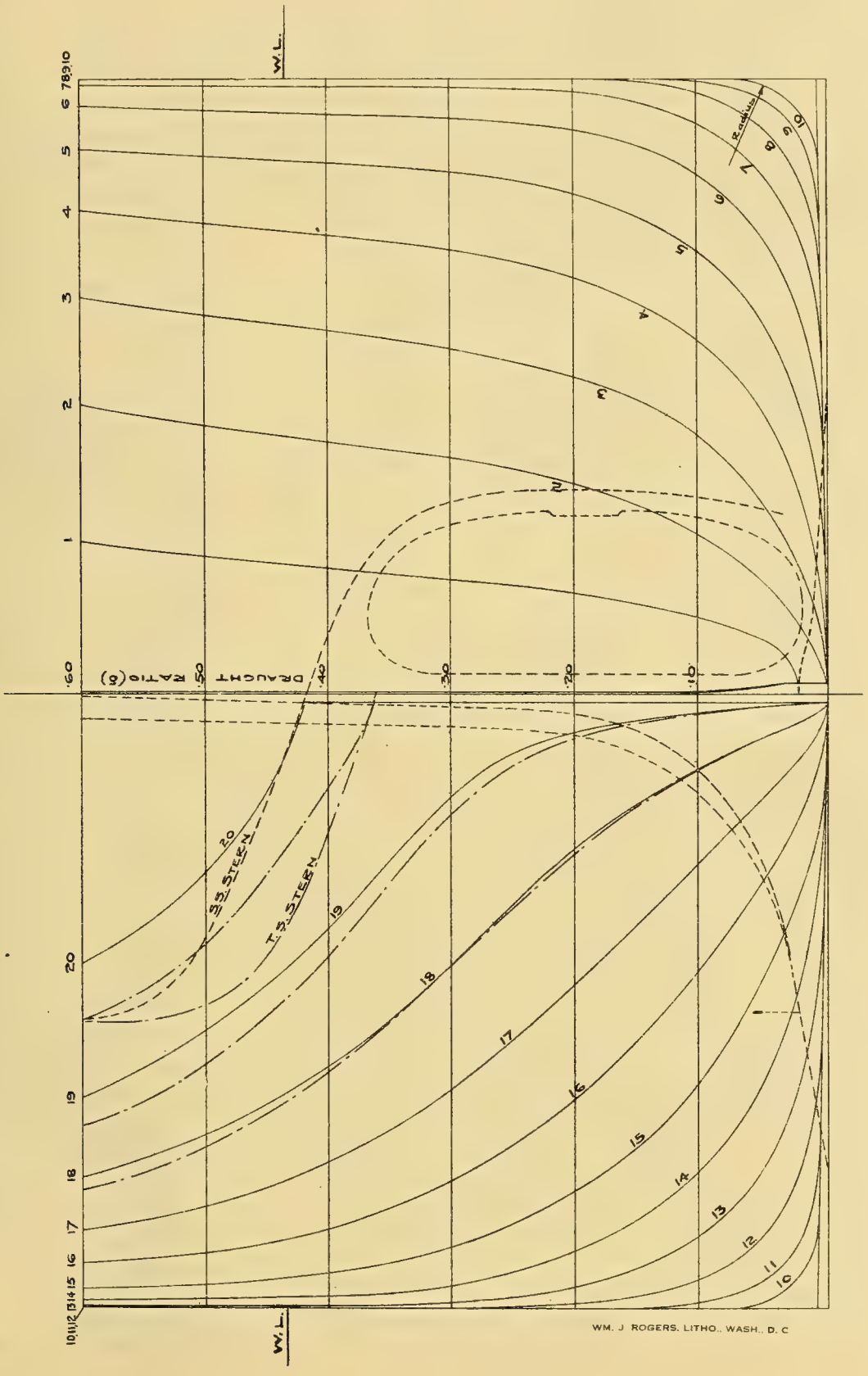


WM. J. ROGERS, LITHO., WASH., D. C.

To illustrate paper on "Economical Cargo Ships—Some Model Experiments." by Alfred J. C. Robertson, Esq., Member.

FIG. 8.



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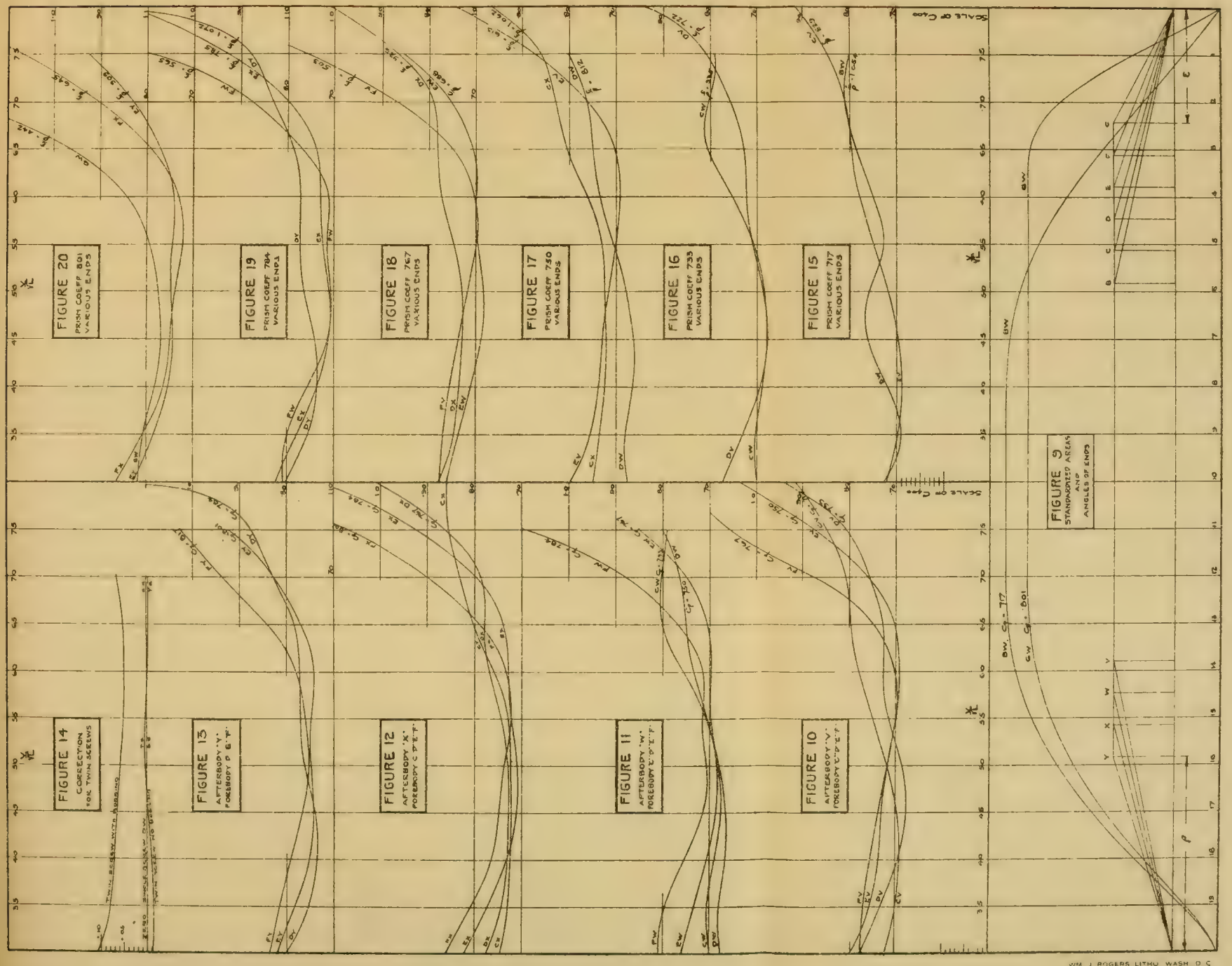

WM J ROGERS LITHO WASH D C

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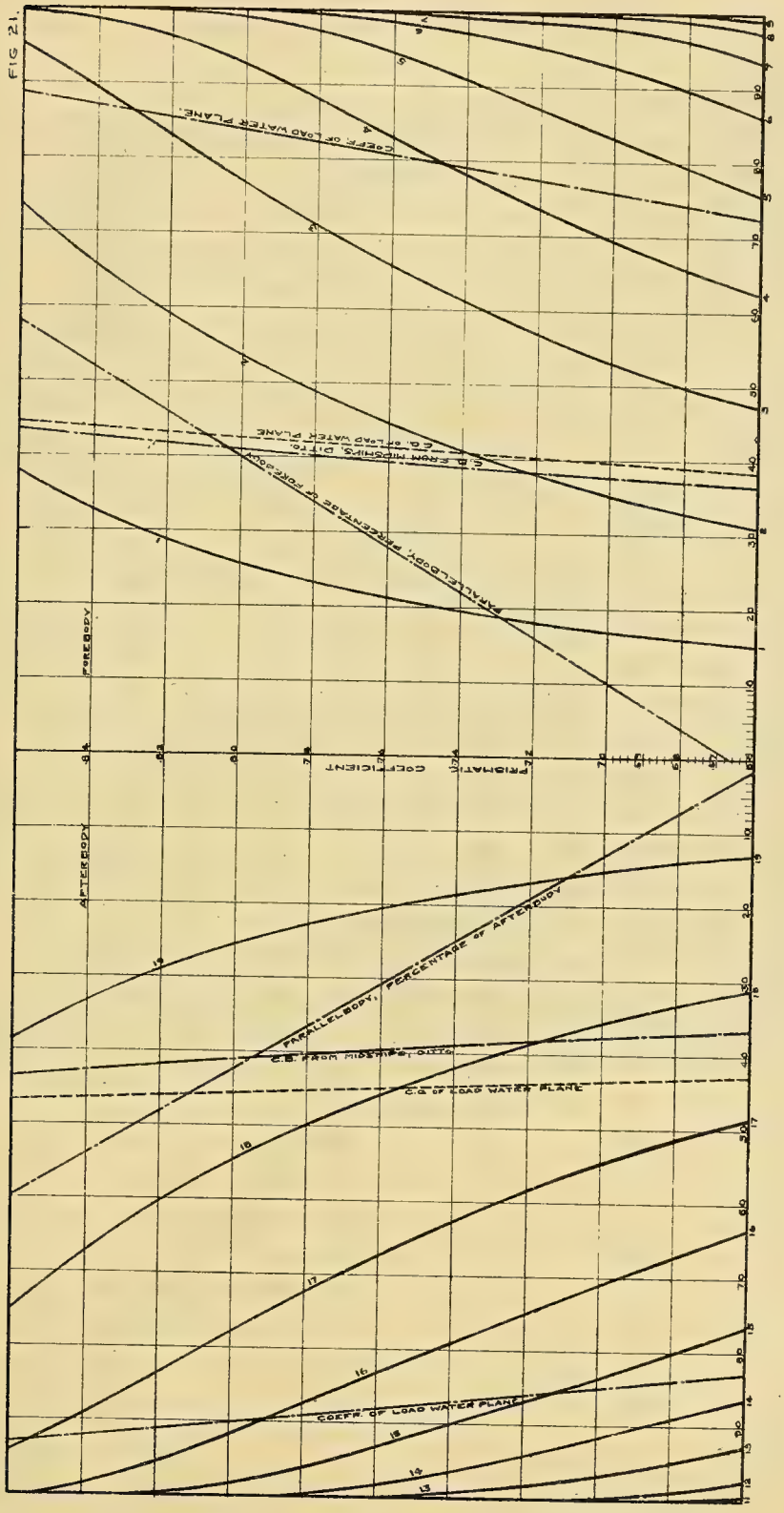

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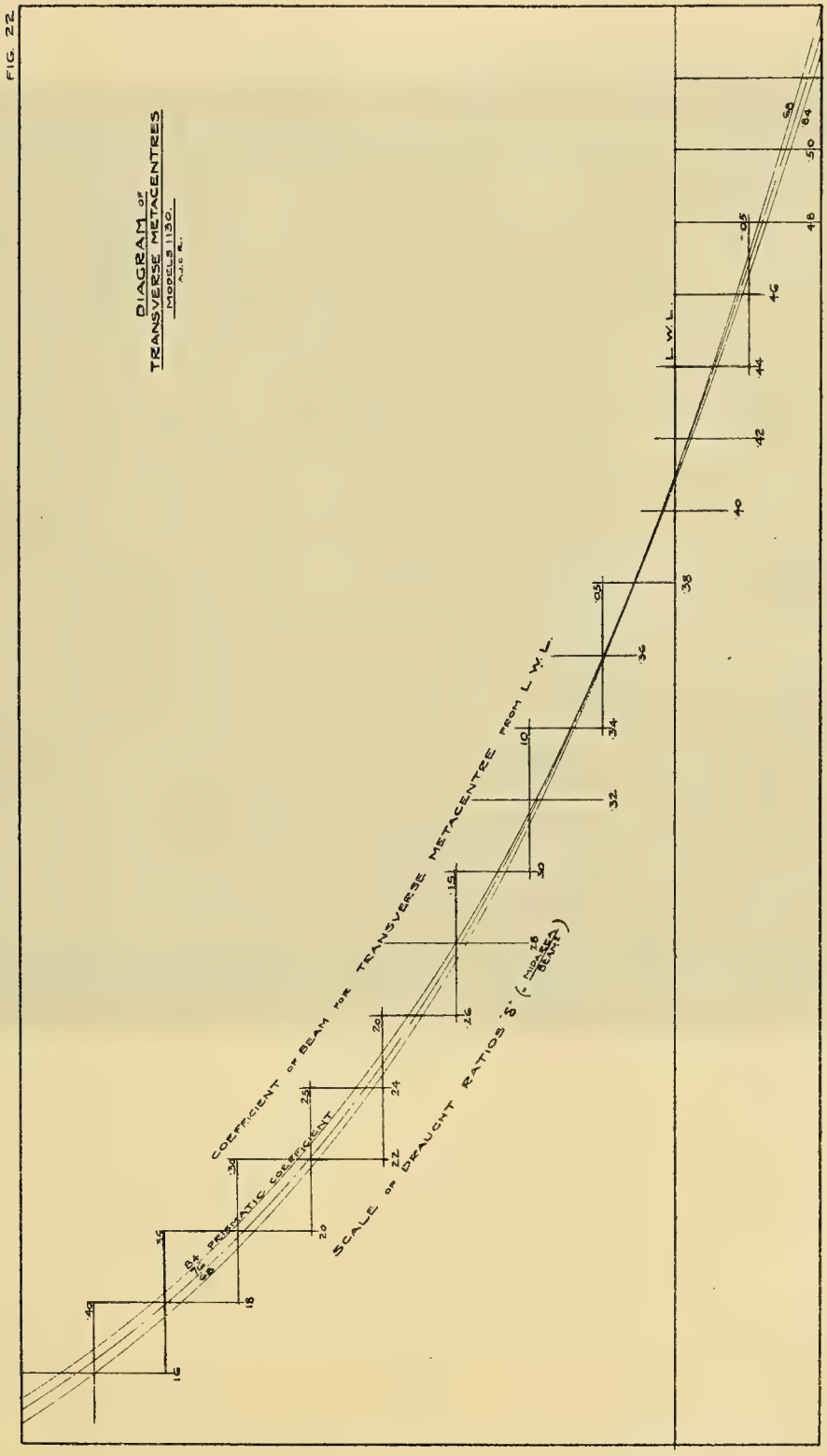

To illustrate discussion by Captain Robert Stocker, C. C., U. S. N., on paper entitled "Notes on Rivets and Spacing of Rivets for Oil-tight Work," by Hugo P. Frear, Esq., Member.

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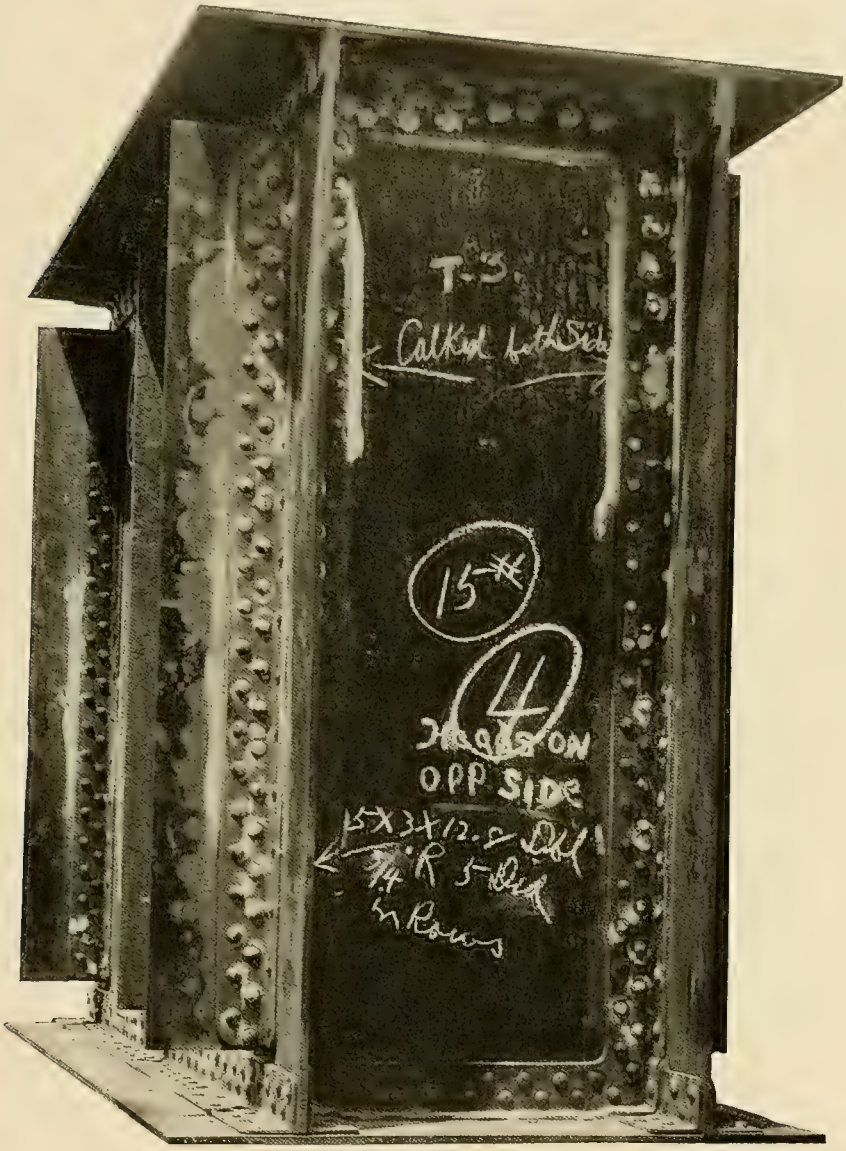

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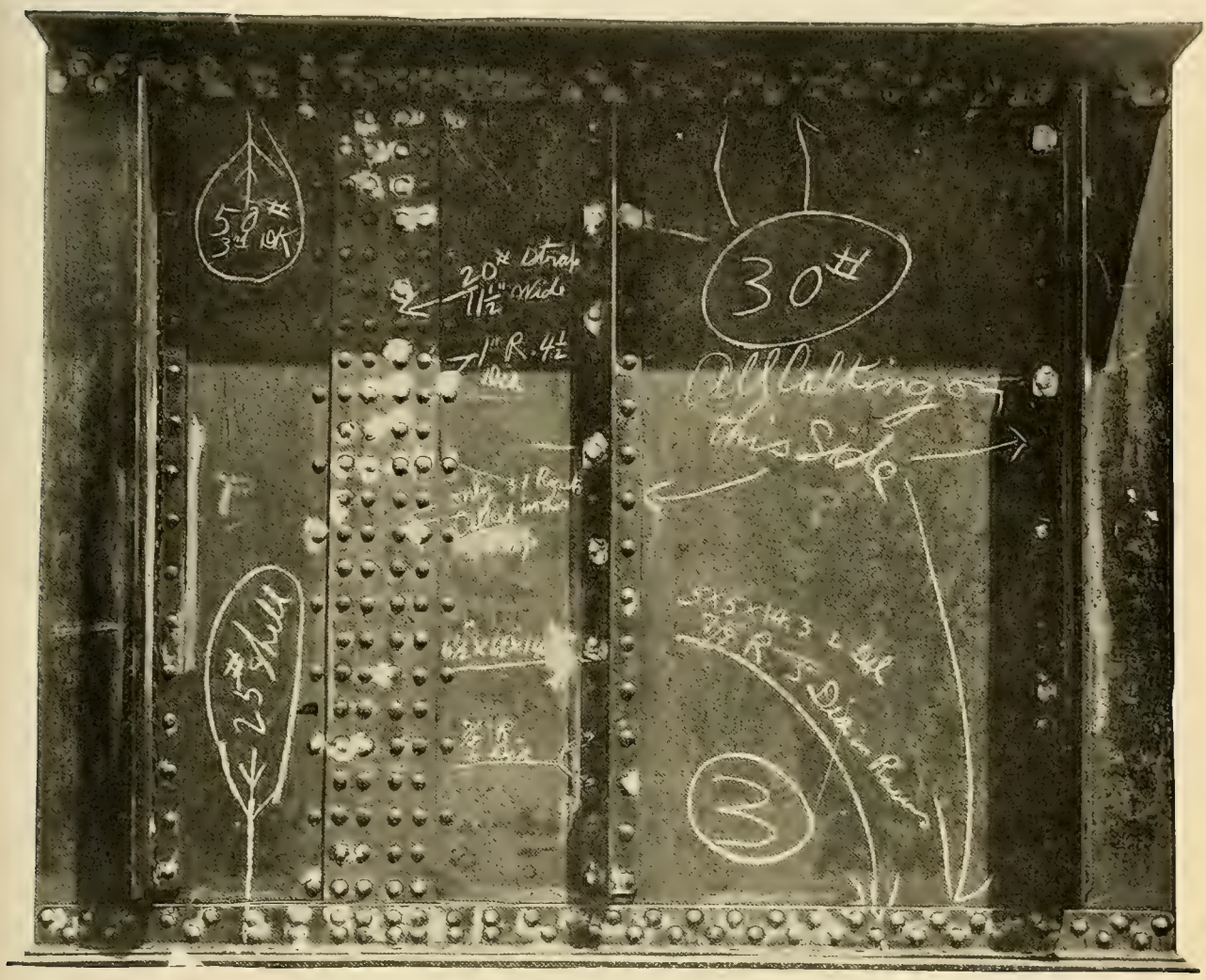

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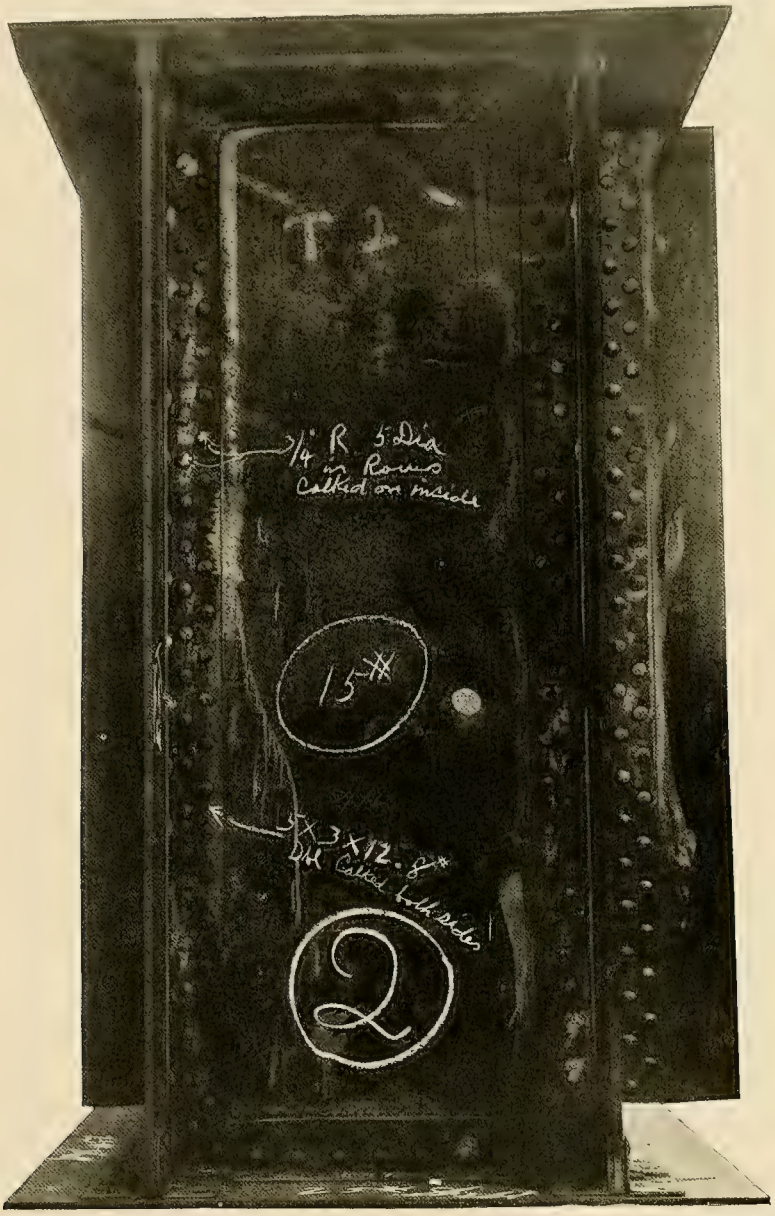

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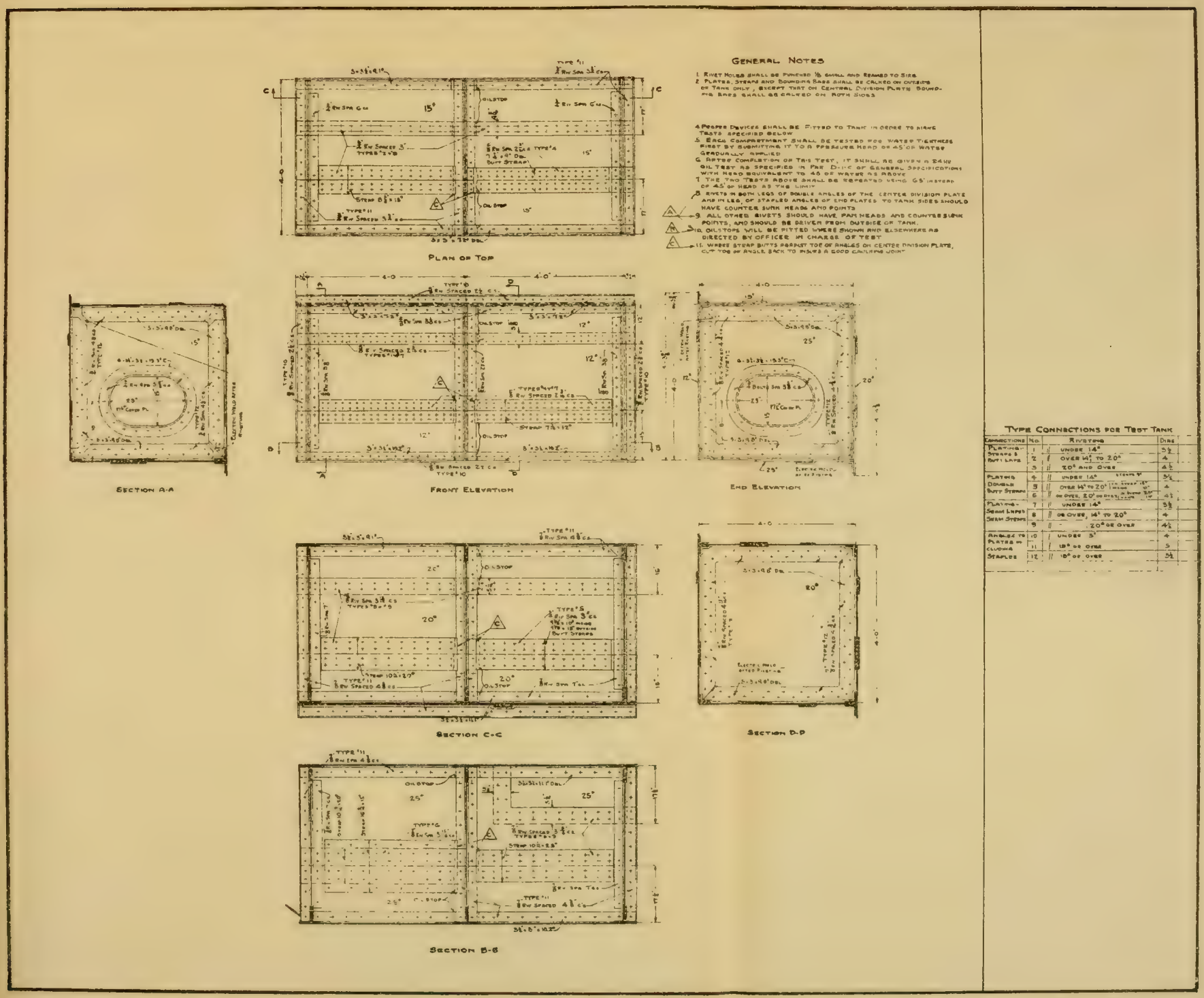
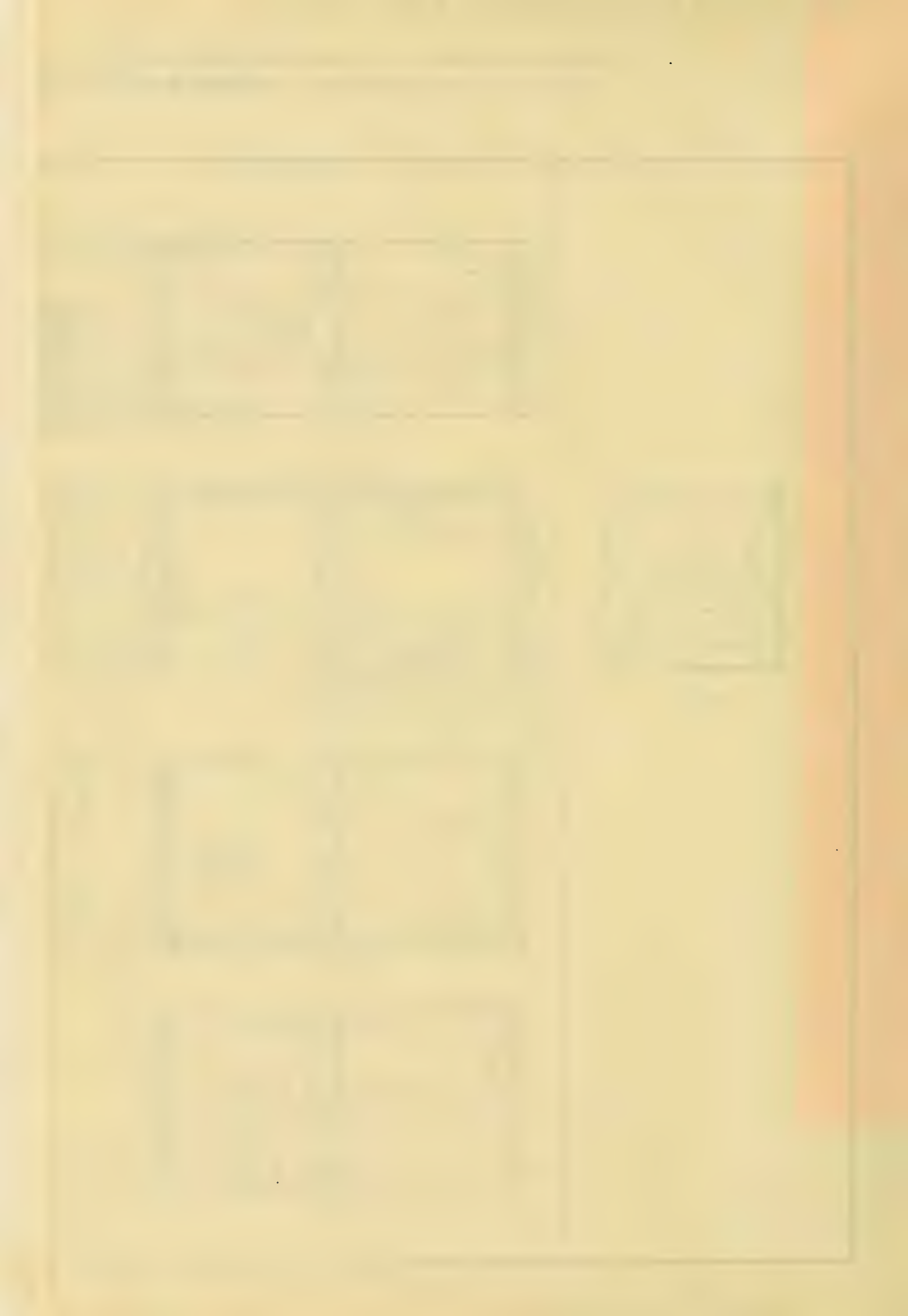


WM. J. ROGERS LITHO. WASH. D. C.



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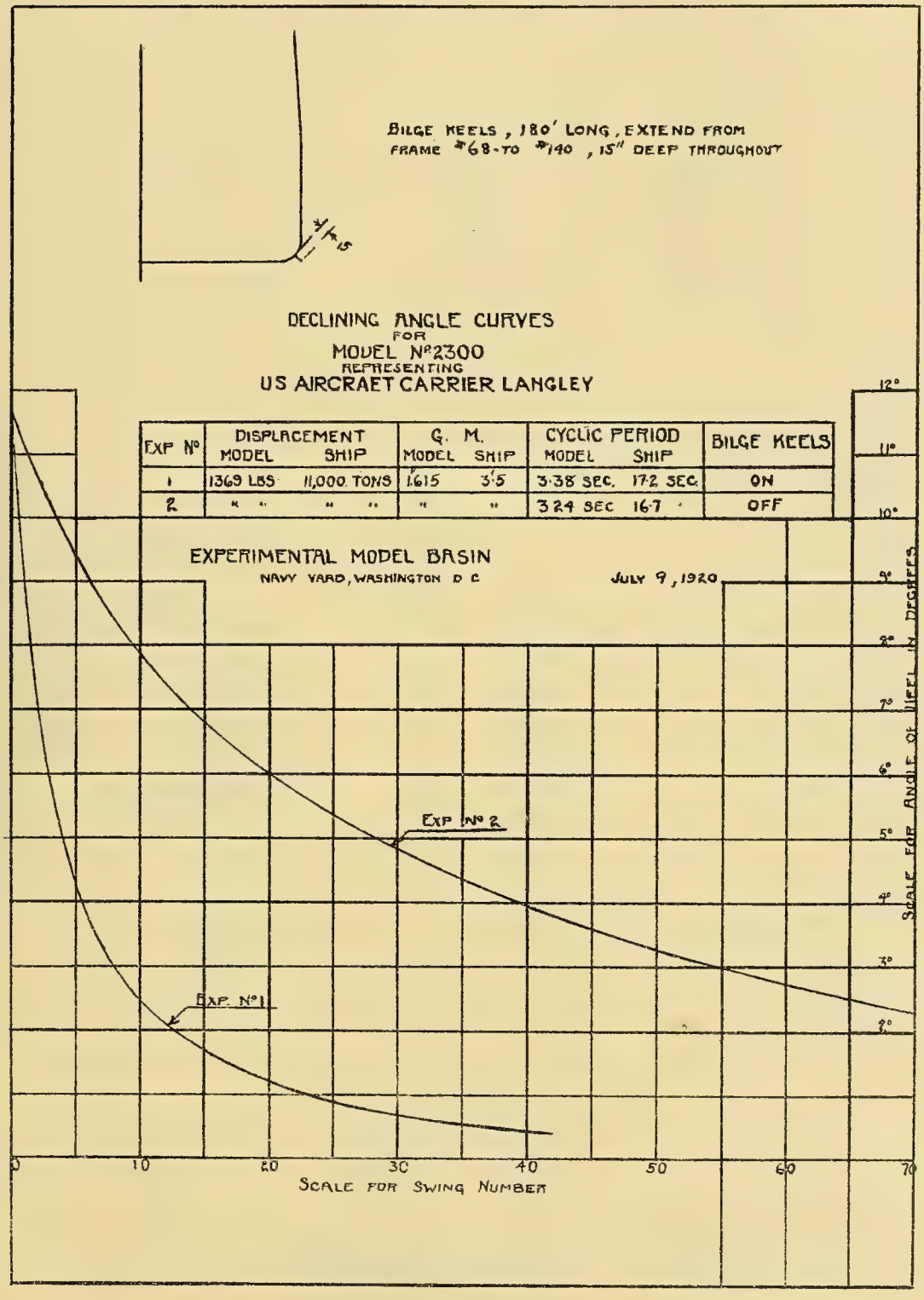

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GYROSCOPIC ROLLING RECORDS
RECORDS FOR DECLINING ANGLE CURVES FOR
MODEL Nº 2300
REPRESENTING
U.S. AIRCRAFT CARRIER LANGLEY

MODEL SHIP
G.M. 1·615 3·5
DISPLACEMENT 1369 LBS 11,000 TONS
LINEAR RATIO, SHIP TO MODEL = 26

EXPERIMENTAL MODEL BASIN
NAVY YARD, WASHINGTON, D.C.
JULY 1920

WM J ROGERS, LITHO., WASH D.C.

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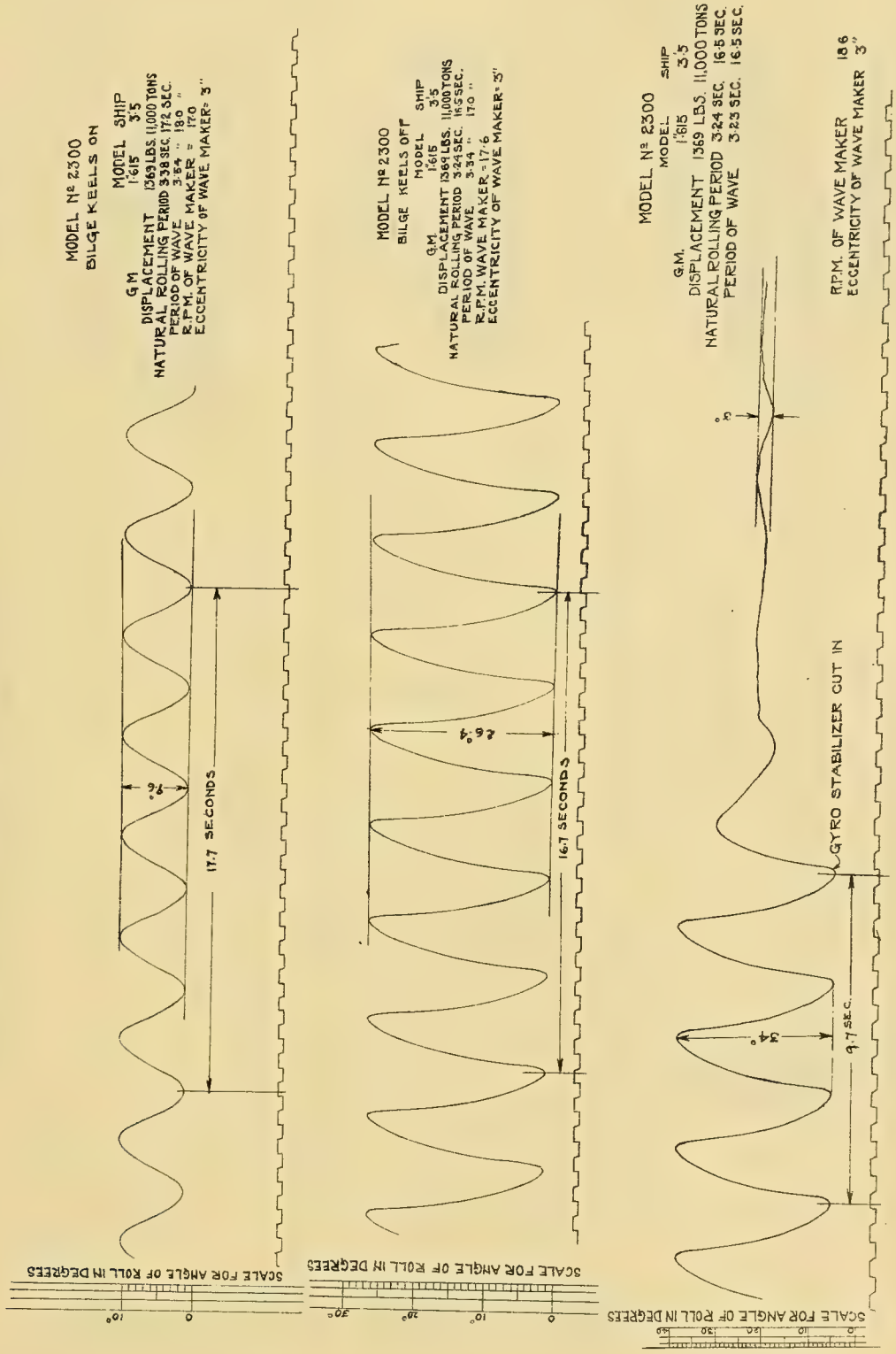

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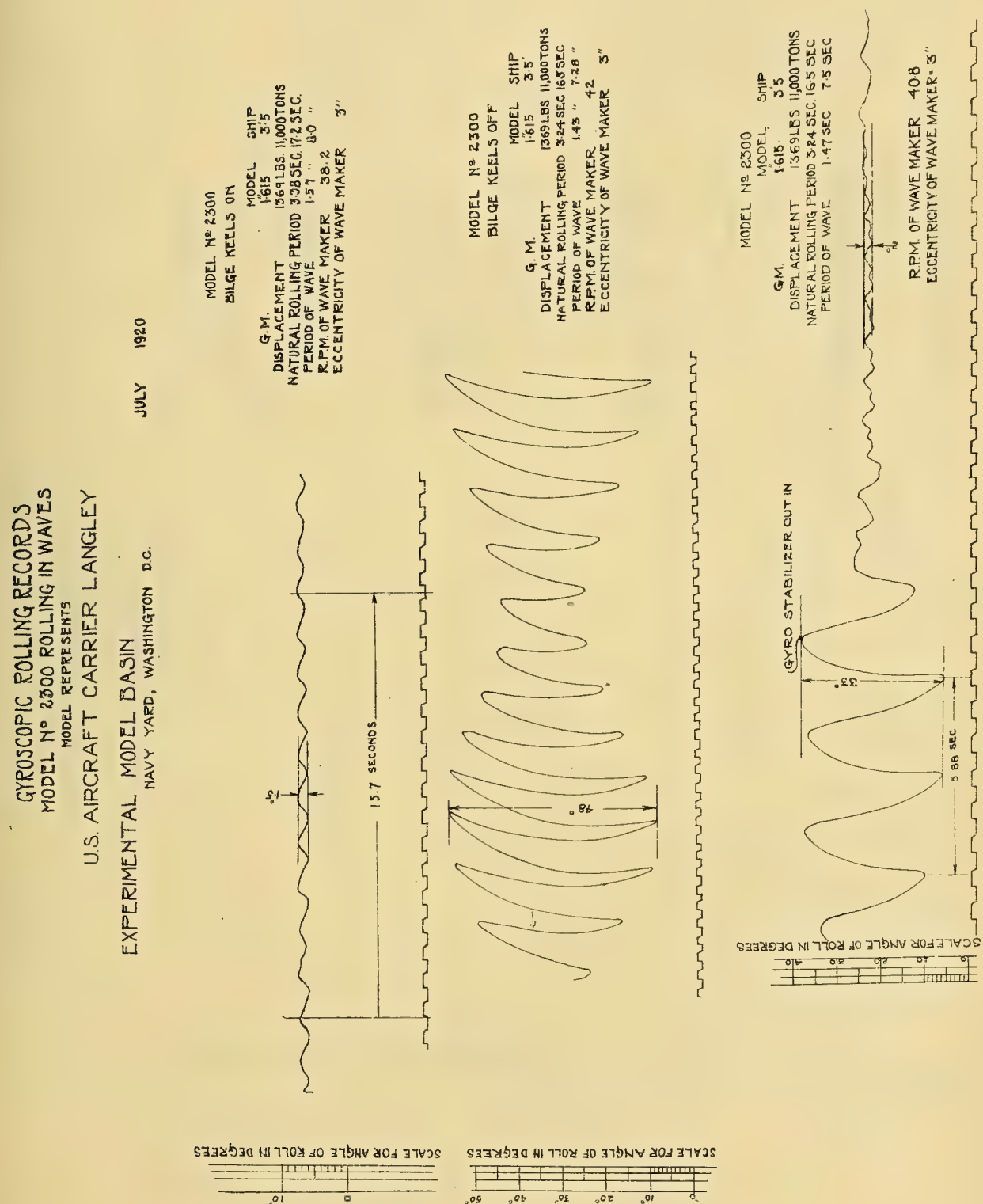

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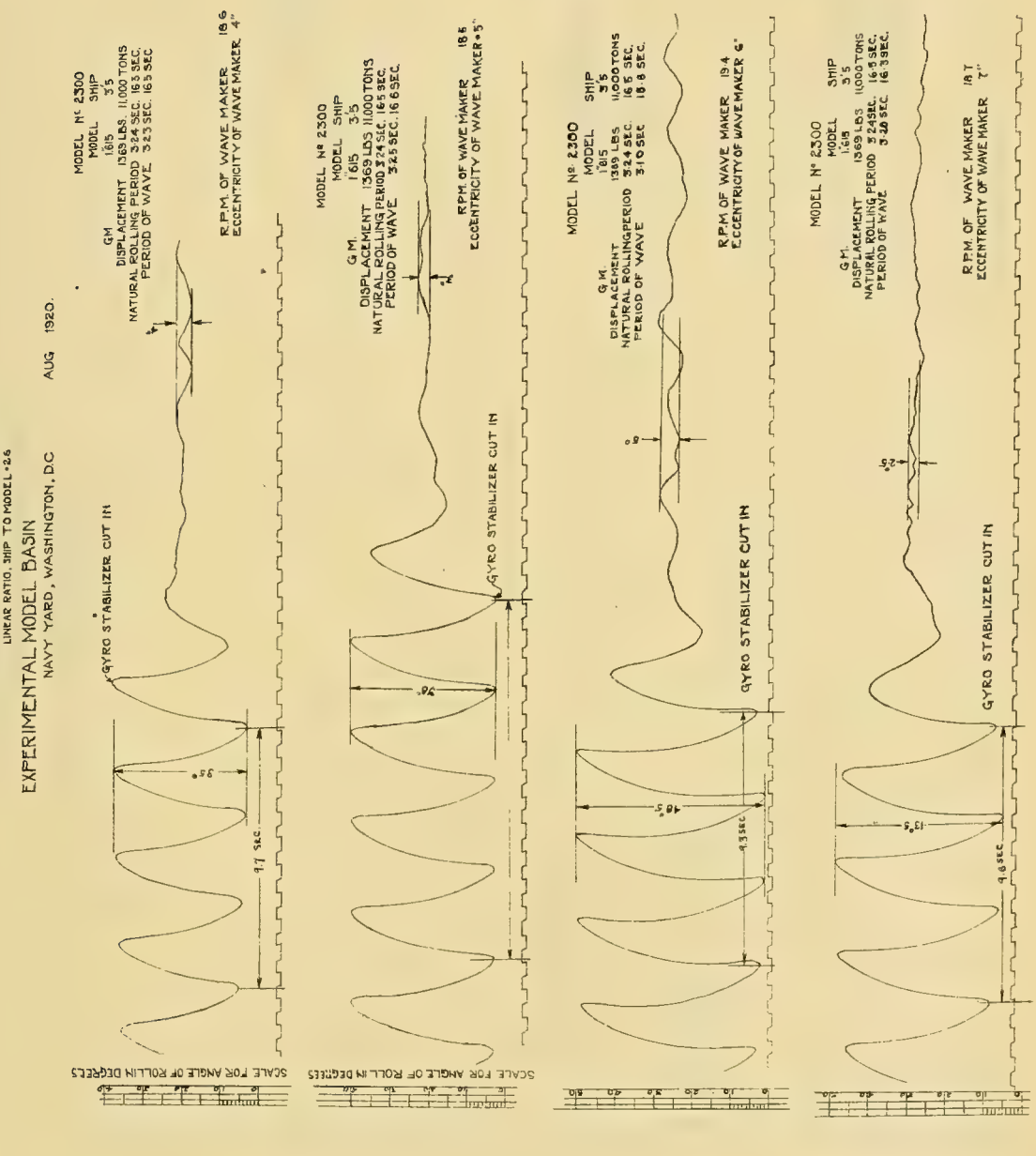
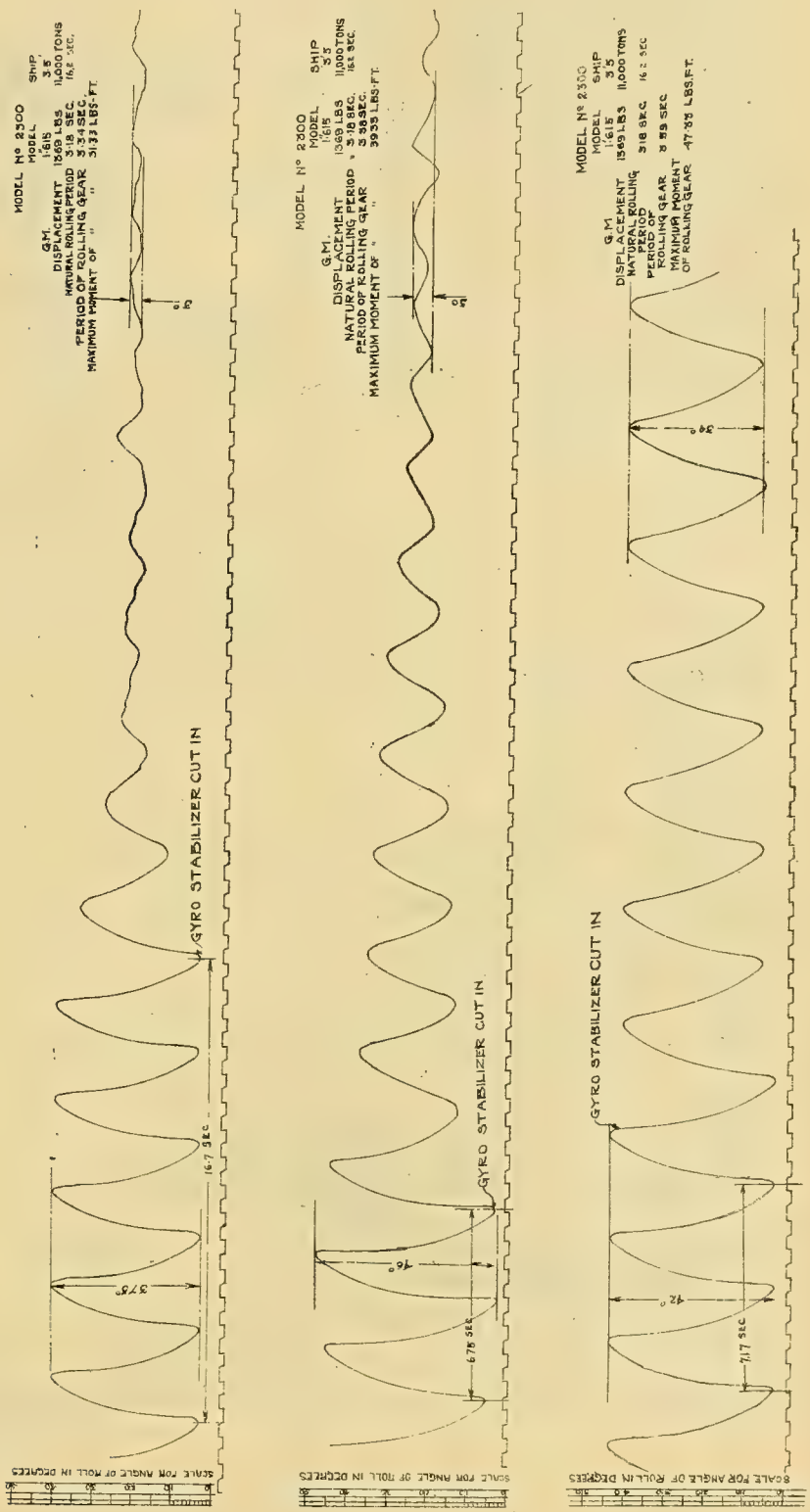


To illustrate paper on "Comparative Tests of Bilge Keels and a Gyro-Stabilizer on a Model of the U. S. Aircraft Carrier Langley," by Commander William McEntee, Construction Corps, U. S. N., Member.

GYROSCOPIC ROLLING RECORDS
MODEL Nº 2300 ROLLED IN STILL WATER
BY WEIGHT SHIFTING GEAR
WITH AND WITHOUT GYRO STABILIZER
MODEL REPRESENTS
U.S. AIRCRAFT CARRIER LANGLEY
LINEAR RATIO SHIP TO MODEL - 48
EXPERIMENTAL MODEL BASIN
NAVY YARD, WASHINGTON D.C. AUG., 1920.



WM. J. ROGERS LITHO. WASH. D. C.

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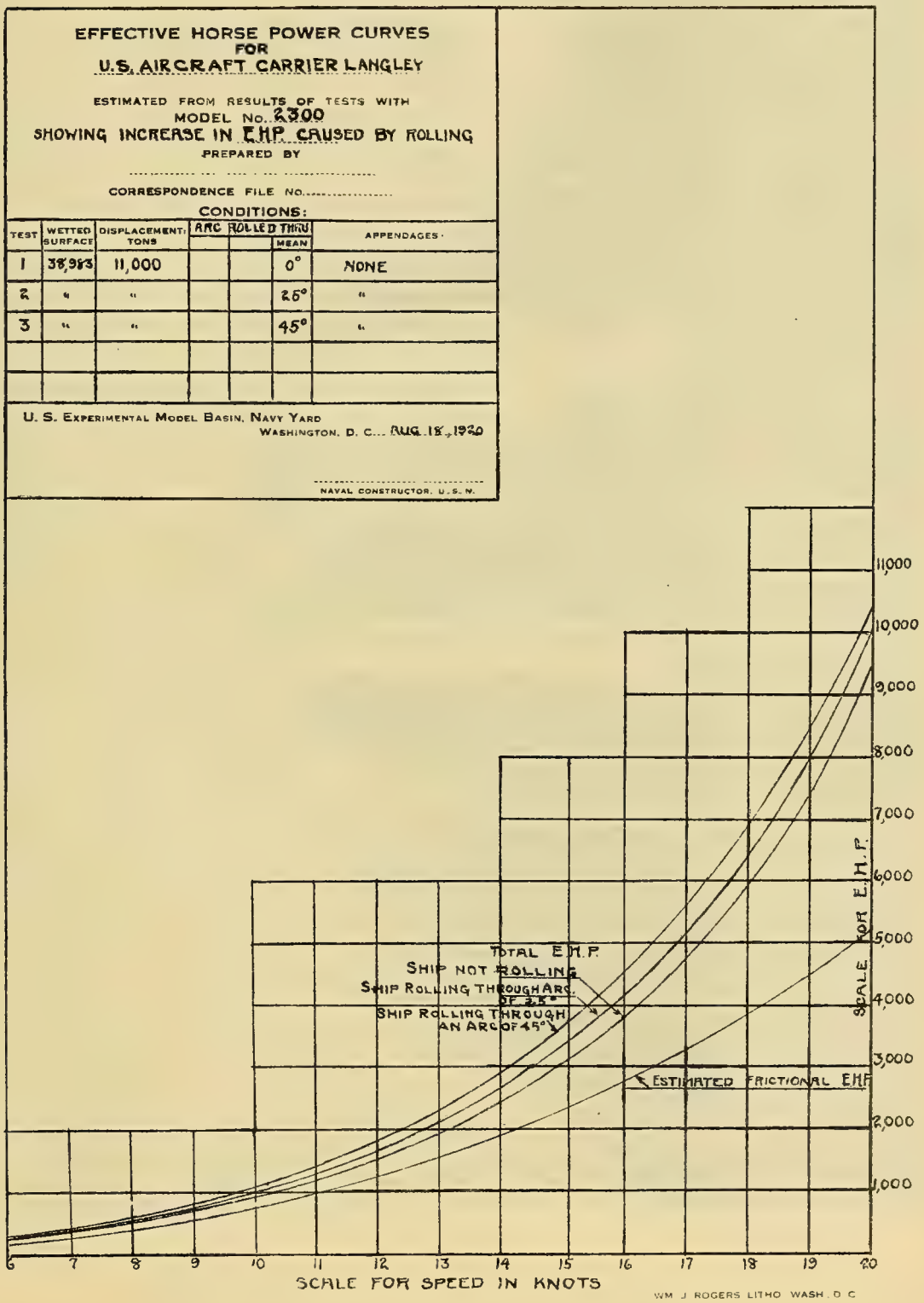

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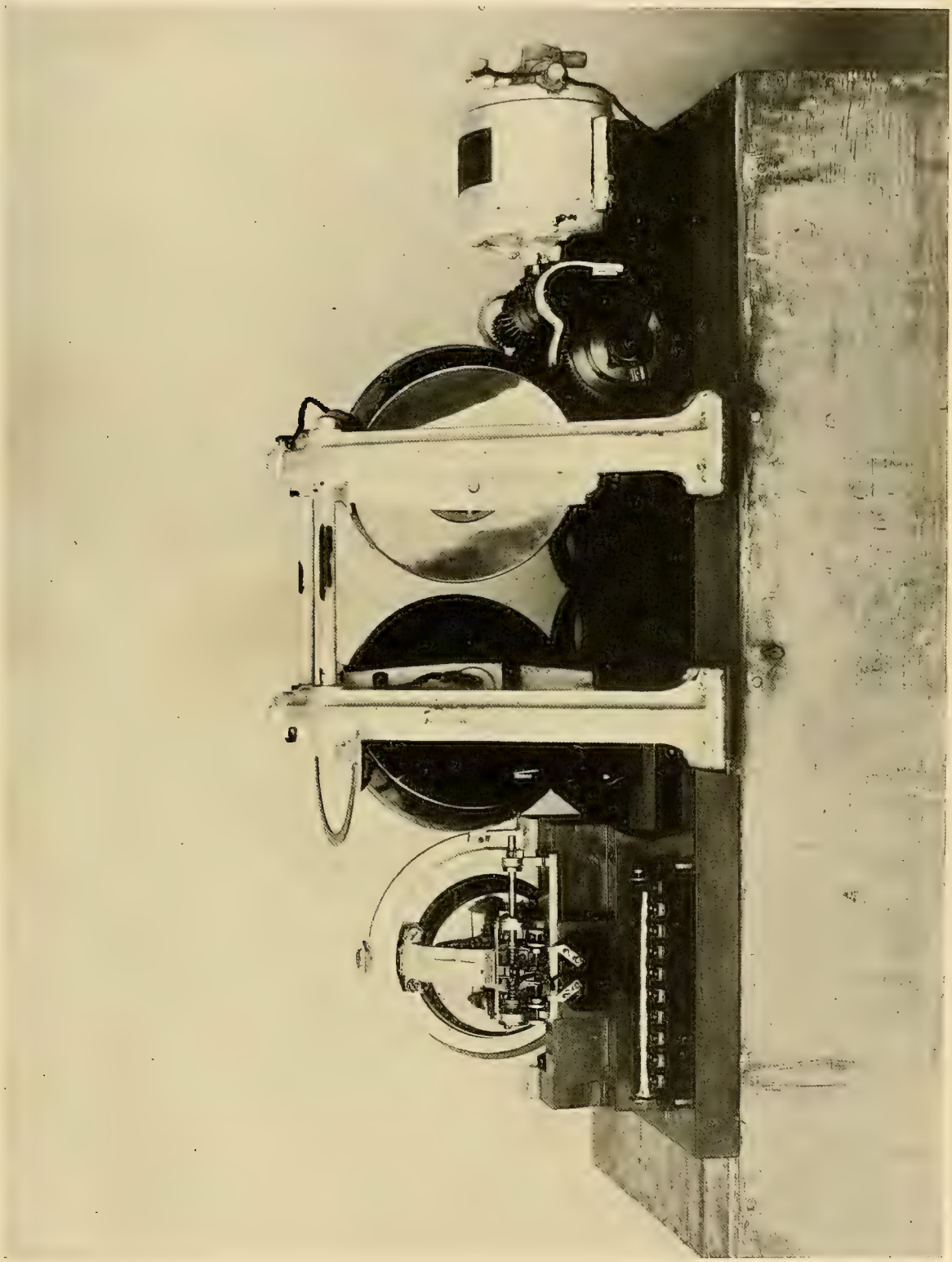
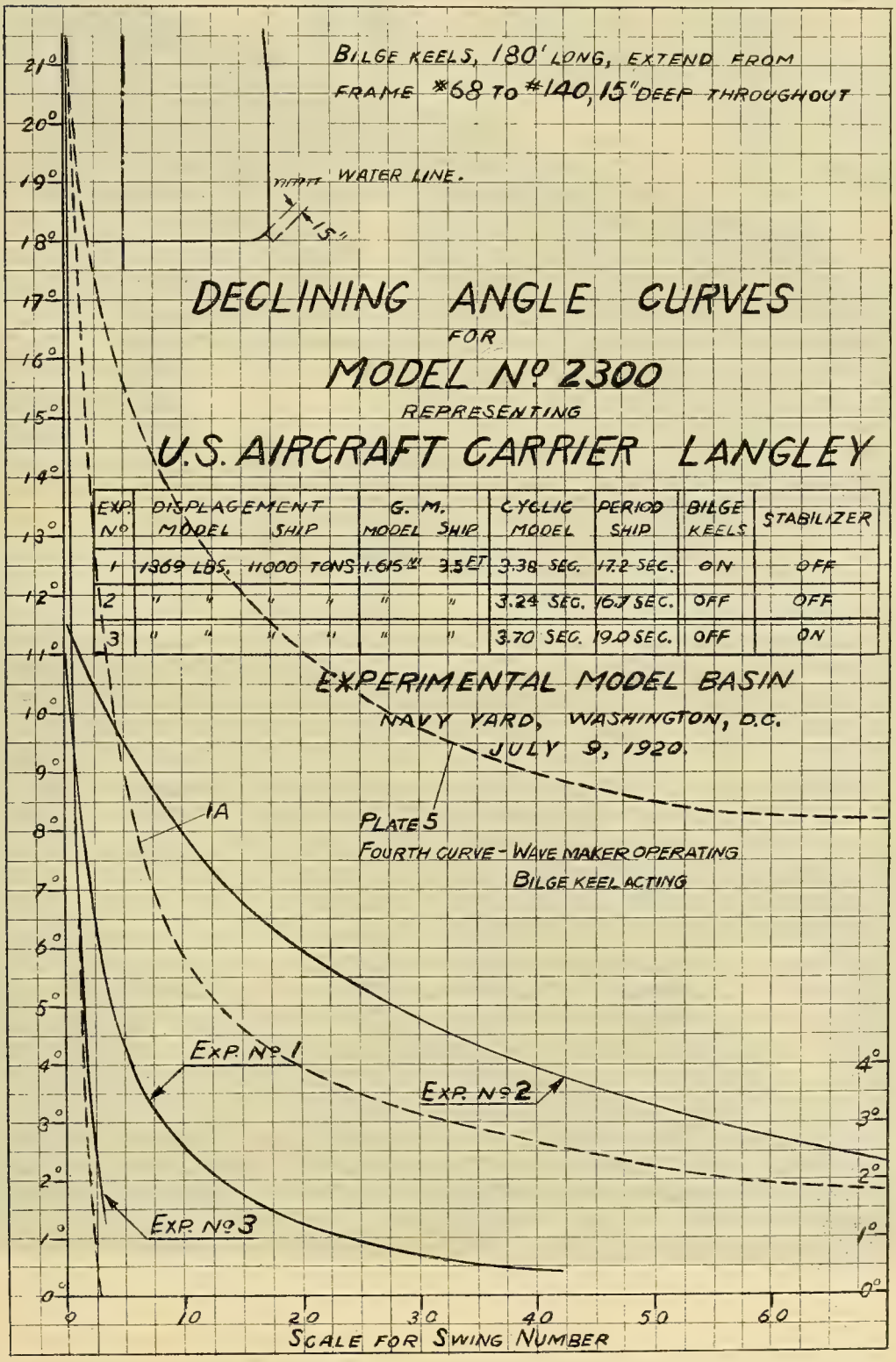


To illustrate discussion by Elmer A. Sperry, Esq., on paper entitled "Comparative Tests of Bilge Keels and a Gyro-Stabilizer on a Model of the U. S. Aircraft Carrier Langley," by Commander William McEntee, Construction Corps, U. S. N., Member.



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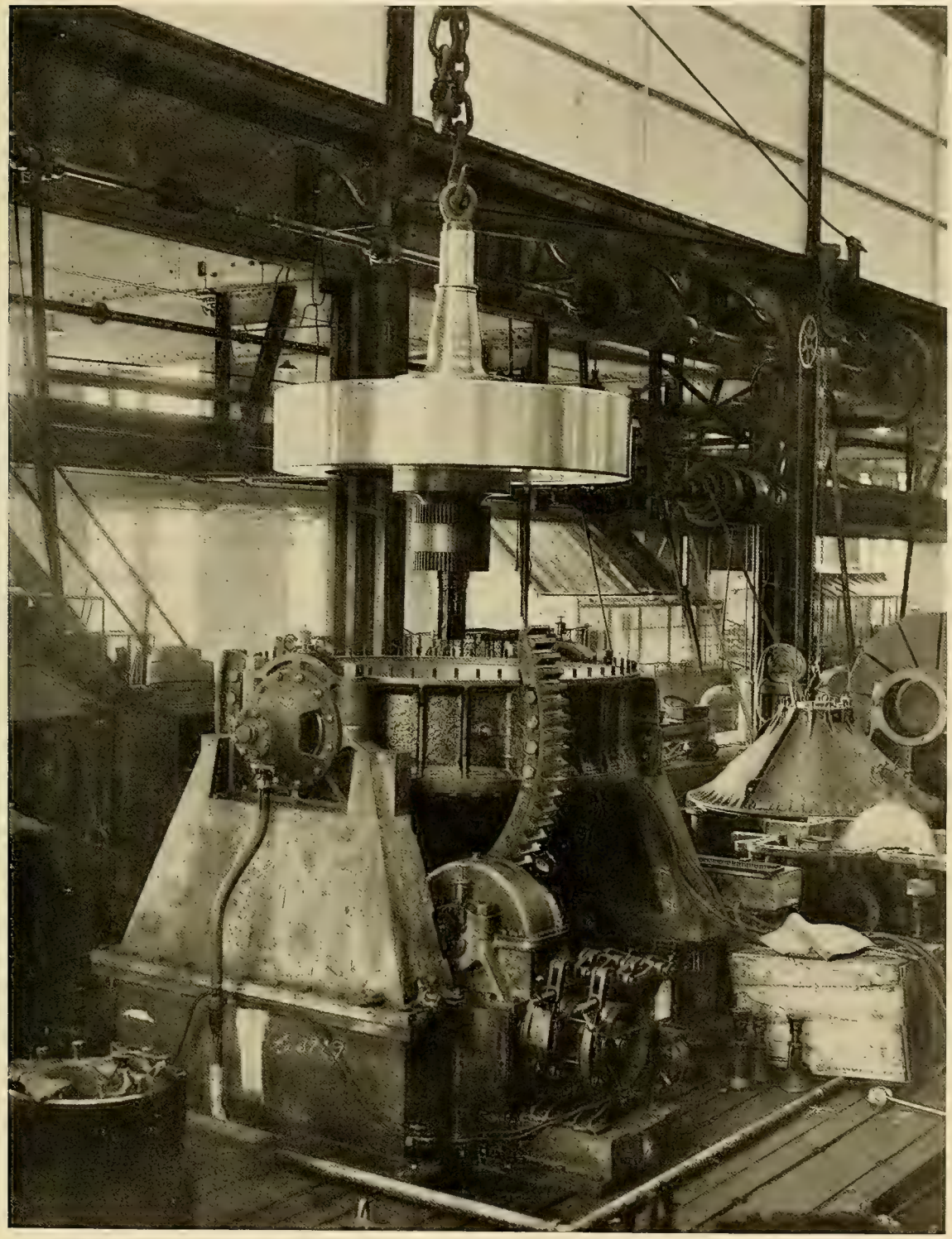
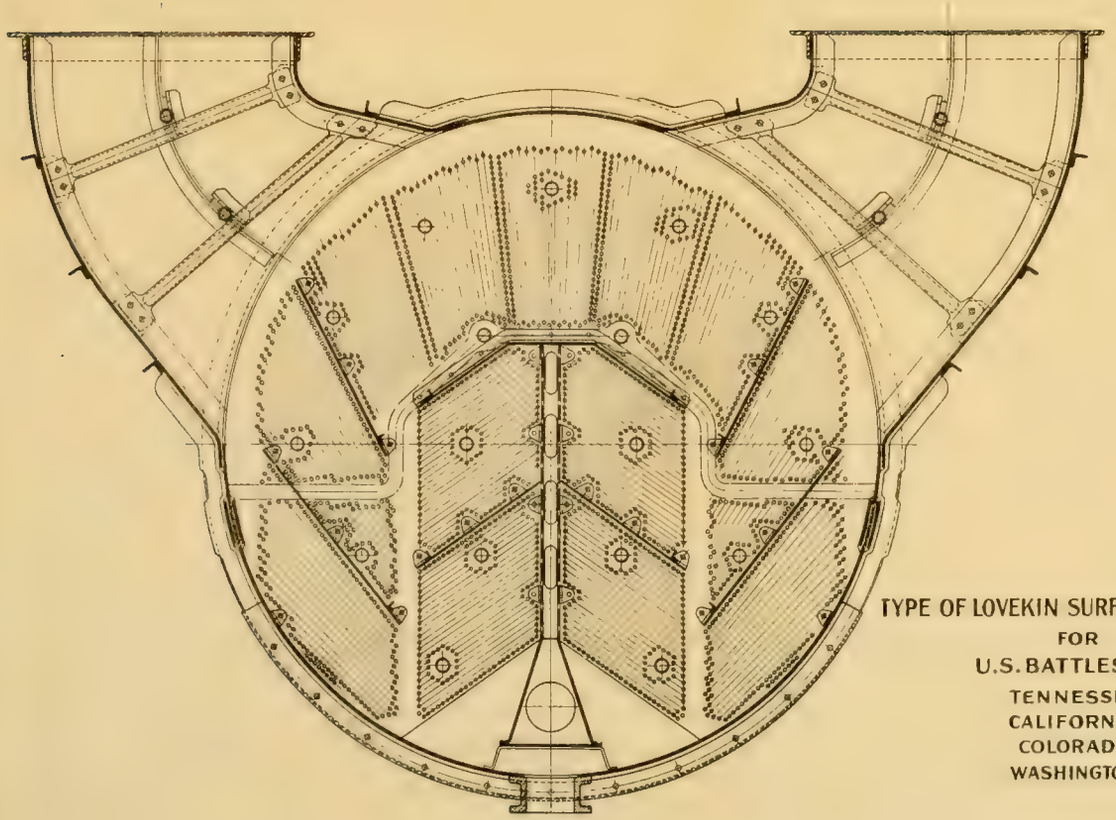
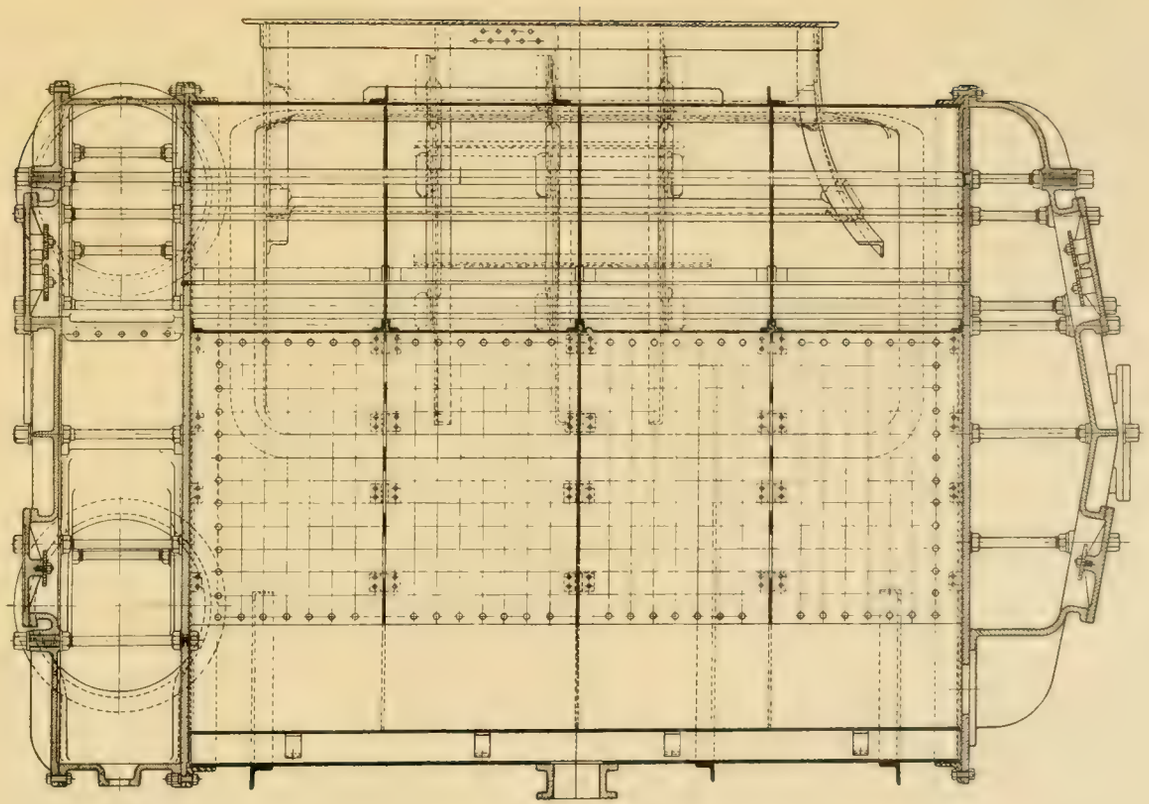


To illustrate paper on "Surface Condensers,"
by Luther D. Lovekin, Esq., Member.



TYPE OF LOVEKIN SURFACE CONDENSER
FOR
U.S. BATTLESHIPS
TENNESSEE
CALIFORNIA
COLORADO
WASHINGTON



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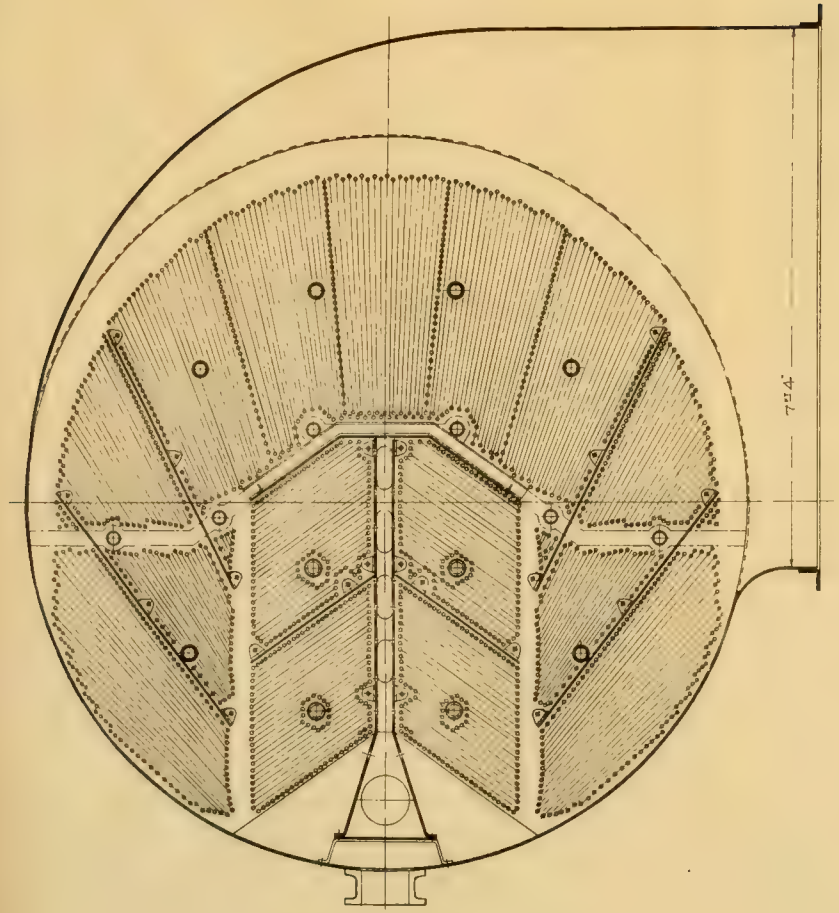
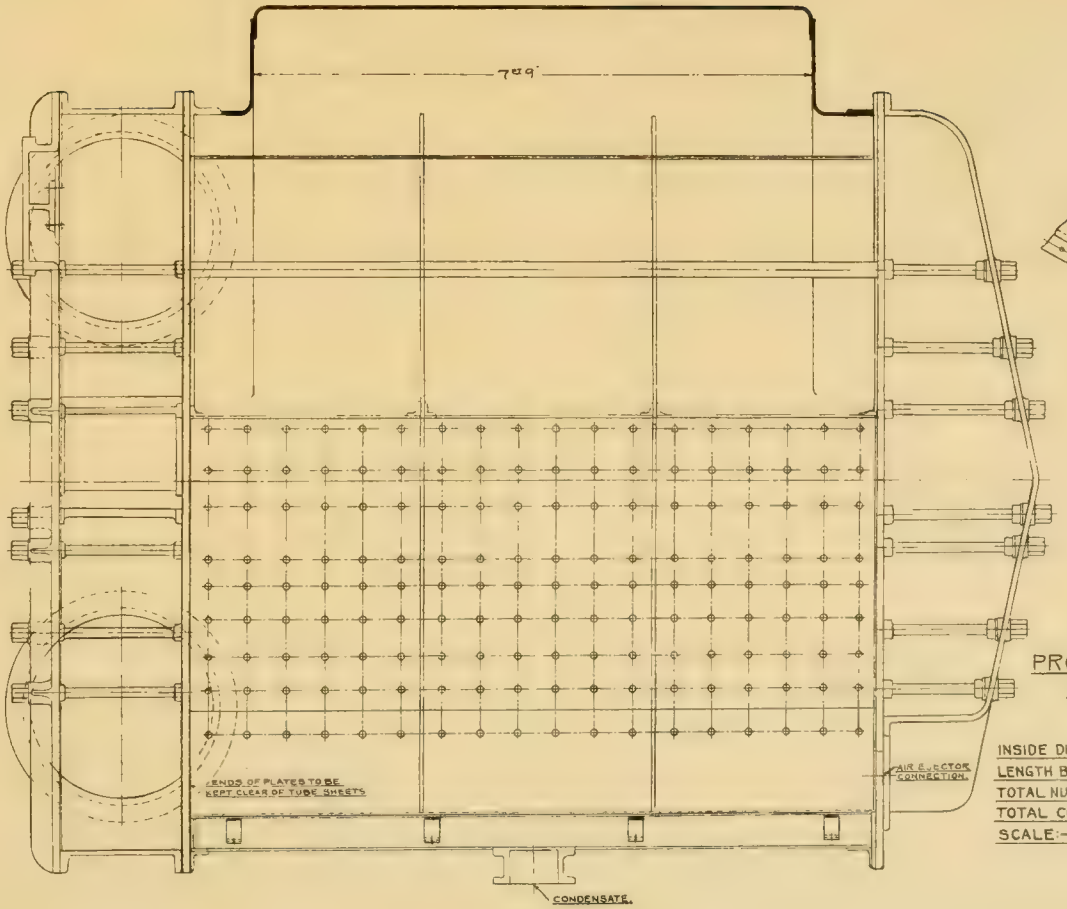
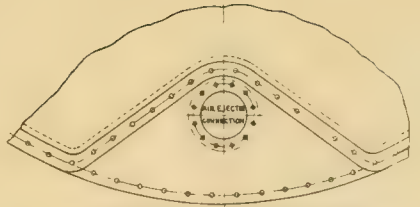




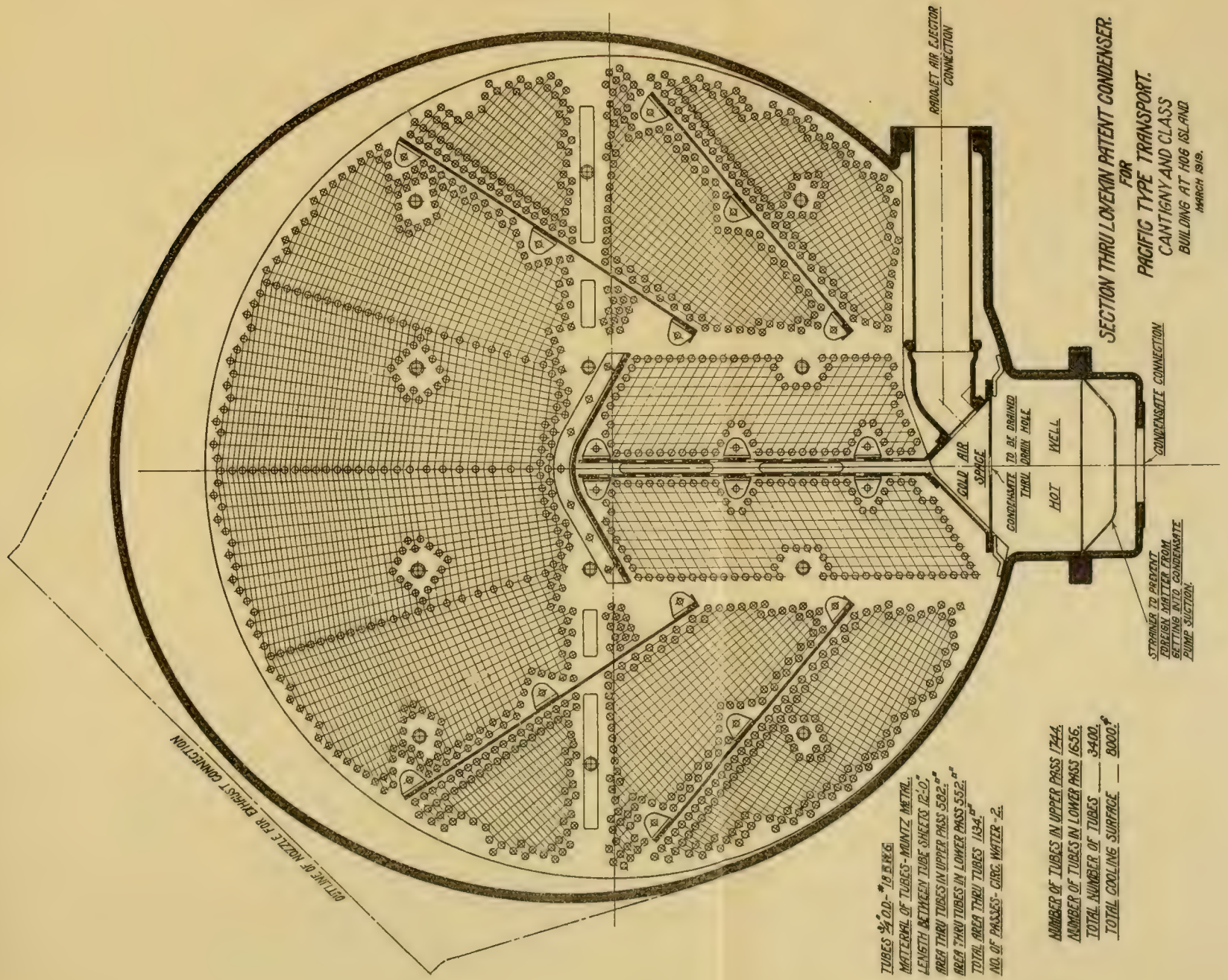


OUTSIDE VIEW OF WATER CHEST
SHOWING AIR EJECTOR CONNECTION

PROPOSED SURFACE CONDENSER
FOR U. S. BATTLE CRUISER.
LOVEKIN PATENT.

INSIDE DIAMETER	10 ft 0
LENGTH BETWEEN TUBE SHEETS	9 ft 6
TOTAL NUMBER OF TUBES ($\frac{5}{8}$ O.D. – 18 B.W.G.)	9000
TOTAL COOLING SURFACE	14000 sq ft
SCALE:– $1\frac{1}{2}$" = 1 ft	DATE:– JULY 2, 1920

To illustrate paper on "Surface Condensers," by Luther D. Lovekin, Esq., Member.



To illustrate paper on "Surface Condensers, by Luther D. Lovekin, Esq., Member.

TEST FOR EFFICIENCY

U. S. A. T. "CANTIGNY"

A. I. S. C.

MACHINERY INSTALLATION AND OUTFITTING DIVISION.

MAY 20TH, 1920.

HULL No. 670.

From Three Hour Readings Taken from Three Hour Full Speed Natural Draft River Trial,
From Two and One Half Hour Readings Taken From Three Hour Full Speed Induced Draft River Trial.
Draft :- Forward, 17'0", Aft, 21'9", Mean 19'4½"
Propeller :- Diam. 18'6", Pitch 17.056 FT

ENGINE ROOM LOG SHEET "C"

TIME.	BAROMETER	VACUUMS. AIR EJECTOR, (RADOJET TYPE)	VACUUMS. DRY SUCTION.	VACUUMS. PORT SIDE, CONDENSER. AFT	PORT SIDE, CONDENSER. CENTER.	PORT SIDE, CONDENSER. FORWARD.	TOP OF CONDENSER. AFT.	TOP OF CONDENSER. CENTER.	TOP OF CONDENSER. FORWARD.	STBD. SIDE, CONDENSER. AFT.	STBD. SIDE, CONDENSER. CENTER	STBD. SIDE, CONDENSER. FORWARD.	VACUUMS. TURBINE EXHAUST.	R.P.M. CIRCULATOR	TEMPERATURE OF COOLING WATER. SEA.	TEMPERATURE OF COOLING WATER. DISCHARGE OVERBOARD	STEAM PRESSURE ON RADOJET.	CONDENSATE	TURBINE EXHAUST. ABSOLUTE PRESSURE	TURBINE EXHAUST. TEMP CORRS. TO ABSOLUTE PRESSURE
									NATURAL DRAFT											
P M																				
12.49.	–		28.87	29.00	28.86	29.03	28.92	28.80	28.80	28.81	29.02	28.81	28.95	–	59.5	78	–	84	–	–
1.19	–	29.4	28.84	28.96	28.82	28.98	28.87	28.85	28.78	28.78	28.98	28.77	29.00	235	59	76	150	82	–	–
1.49	30.07	29.6	28.87	29.00	28.85	29.02	28.90	28.89	28.80	28.82	29.00	28.80	29.00	245	58.5	76	150	83	1.07	–
2.19	30.07	29.5	28.90	29.02	28.87	29.00	28.92	28.90	28.82	28.84	29.01	28.81	28.95	250	58.5	76	150	84	1.12	–
2.49	30.09	29.6	28.85	28.98	28.85	28.90	28.90	28.90	28.80	28.83	29.02	28.80	28.85	240	58	77	155	83	1.24	–
3.19	30.09	29.6	28.85	28.98	28.85	28.96	28.92	28.93	28.82	28.85	29.00	28.80	28.95	250	57	76	155	82	1.14	–
3.49	30.08	29.6	28.90	29.01	28.91	29.05	28.95	28.93	28.84	28.83	29.03	28.84	28.97	250	57.5	76	155	82	1.11	–
AVG	30.08	29.55	28.87	28.99	28.86	28.99	28.91	28.90	28.81	28.82	29.01	28.80	28.95	245	58.3	76.6	152.5	82.9	1.136	83.0
CORR.			+0.12	-0.07	+0.10	-0.07	+0.03	+0.05	+0.16	+0.13	-0.09	+0.10								
CORR. AVG.	30.08	29.55	28.99	28.92	28.96	28.92	28.94	28.95	28.96	28.95	28.92	28.90	28.95	245	58.3	76.6	152.5	82.9	1.136	83.0
									INDUCED DRAFT.											
4.19	30.08	29.6	28.88	29.00	28.92	29.08	28.92	28.90	28.82	28.80	29.08	28.89	28.98	245	57	76.5	155	82	1.10	–
4.49	30.03	29.7	28.85	28.98	28.80	29.00	28.86	28.88	28.80	28.79	29.00	28.81	28.90	255	57.5	77	155	84	1.13	–
5.19	30.02	29.7	28.87	29.01	28.86	29.04	28.90	28.90	28.80	28.81	29.02	28.81	28.90	250	58	76	152	82	1.12	–
5.49	30.02	29.6	28.89	29.04	28.88	29.05	28.92	28.91	28.82	28.83	29.03	28.83	28.92	275	59.5	77	155	84	1.10	–
6.19	30.02	29.6	28.80	28.92	28.80	29.02	28.82	28.85	28.78	28.76	29.00	28.80	28.97	275	60	78.5	150	83	1.05	–
6.49	30.02	29.6	28.79	28.90	28.88	29.00	28.81	28.80	28.73	28.72	28.99	28.80	28.89	265	60	78	156	81	1.13	–
AVG.	30.03	29.63	28.85	28.96	28.87	29.03	28.87	28.87	28.79	28.79	29.02	28.82	28.93	260.8	58.7	77.2	153.7	82.7	1.121	82.5
CORR.			+0.12	-0.07	+0.10	-0.07	+0.03	+0.05	+0.16	+0.13	-0.09	+0.10								
CORR. AVG.	30.03	29.63	28.97	28.91	28.97	28.96	28.90	28.92	28.95	28.92	28.93	28.92	28.93	260.8	58.7	77.2	153.7	82.7	1.121	82.5

For Main Engine Room Log see Sheet "A".

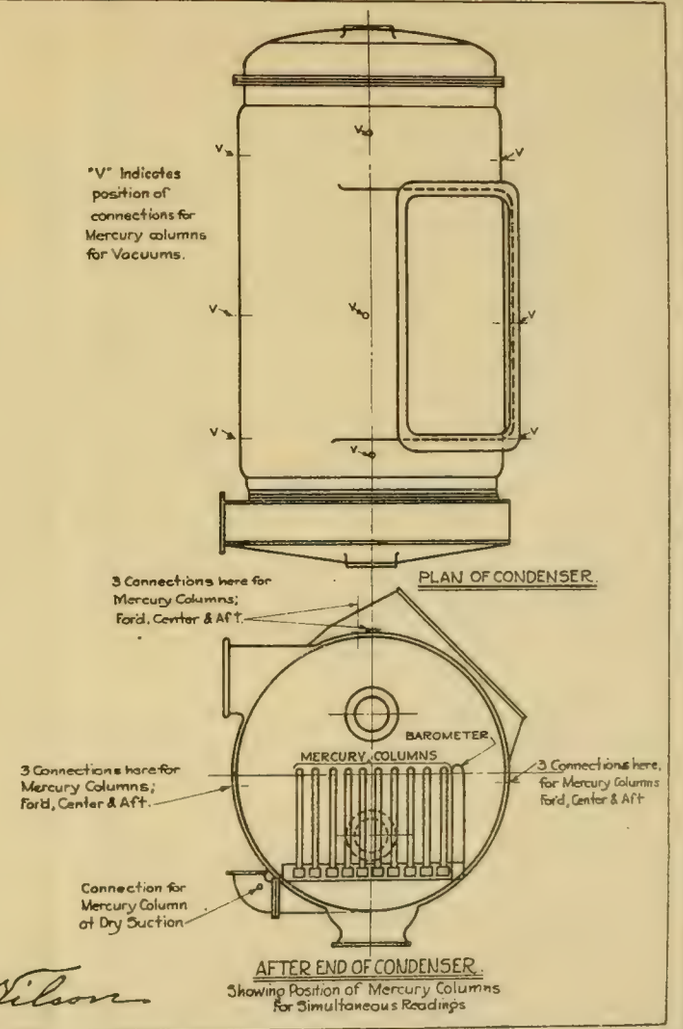
For Lubrication Oil Data, and Turbine Bearing Temperatures, see Sheet "B"

For Fire Room Log, See Sheet "D".

Forward Radojet, only, was used throughout the trial.

The Overboard Discharge Temperature was taken near the Overboard Delivery Valve, so as to insure a thorough mixing of the water.
Condenser was cooling all exhaust steam from a G.E. Turbine, rated 6000 S.H.P., with steam consumption, of 12 lbs. per S.H.P., equalling 72,000 lbs. of steam condensed per hour.
Assuming that Circulating Pump delivered 7.774 gallons of water per minute, 54 lbs. of circulating water condensed One Pound of Steam.

SIGNED BY :- J. A. Wilson



MMS W-586

To illustrate paper on "Surface Condensers," by Luther D. Lovekin, Esq., Member.

June 24, 1920.

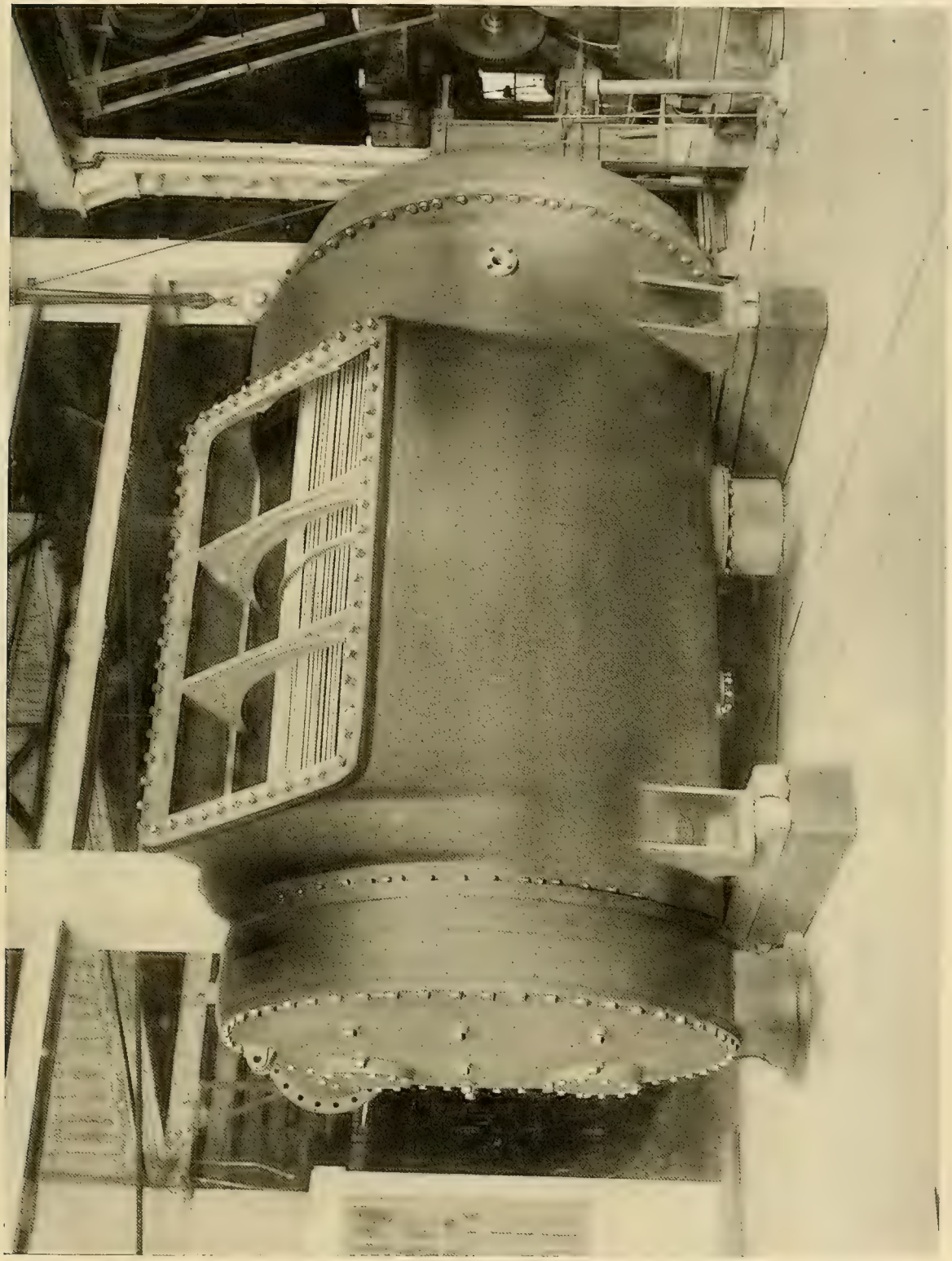
DATA ON CONDENSER ON "B" SHIP "CATIGNY"

NO.	ITEM	UNIT		
1	Run No.		1	2
2	Barometer	in Hg.	30.08	30.03
3	Avg. Vacuum in shell	" "	28.94	28.93
4	Absolute press in shell	" "	1.14	1.10
5	Corresponding temp. ts	Deg. f.	83.2	82.1
6	Inlet water temp. ti	" "	58.3	58.7
7	Discharge water temp. to	" "	76.6	77.2
8	ts - ti	" "	24.9	23.4
9	ts - to	" "	6.6	4.9
10	(ts-ti) ÷ (ts-to)		3.7727	4.7755
11	log e (ts-ti) ÷ (ts-to)		1.3279	1.5637
12	(ts-ti) - (ts-to) = T.R.		18.3	18.5
13	Mean temp. Diff. $\frac{\text{Item 12}}{\text{Item 11}}$		13.78	11.83
14	Surface, outside	sq.ft.	8011	8011
15	Steam condensed	lb/hr	72000	72000
16	Circulating water	lb/hr	3,888,200	3,888,200
17	Heat per lb. steam	Btu	988	999
18	$K = \frac{\text{Btu per hour}}{\text{sq.ft. x deg.}} = \frac{(17) \times (15)}{TD\ (14 \times (13)}$		645	759

Velocity of circulating water thru tubes = 4.37 feet per second.

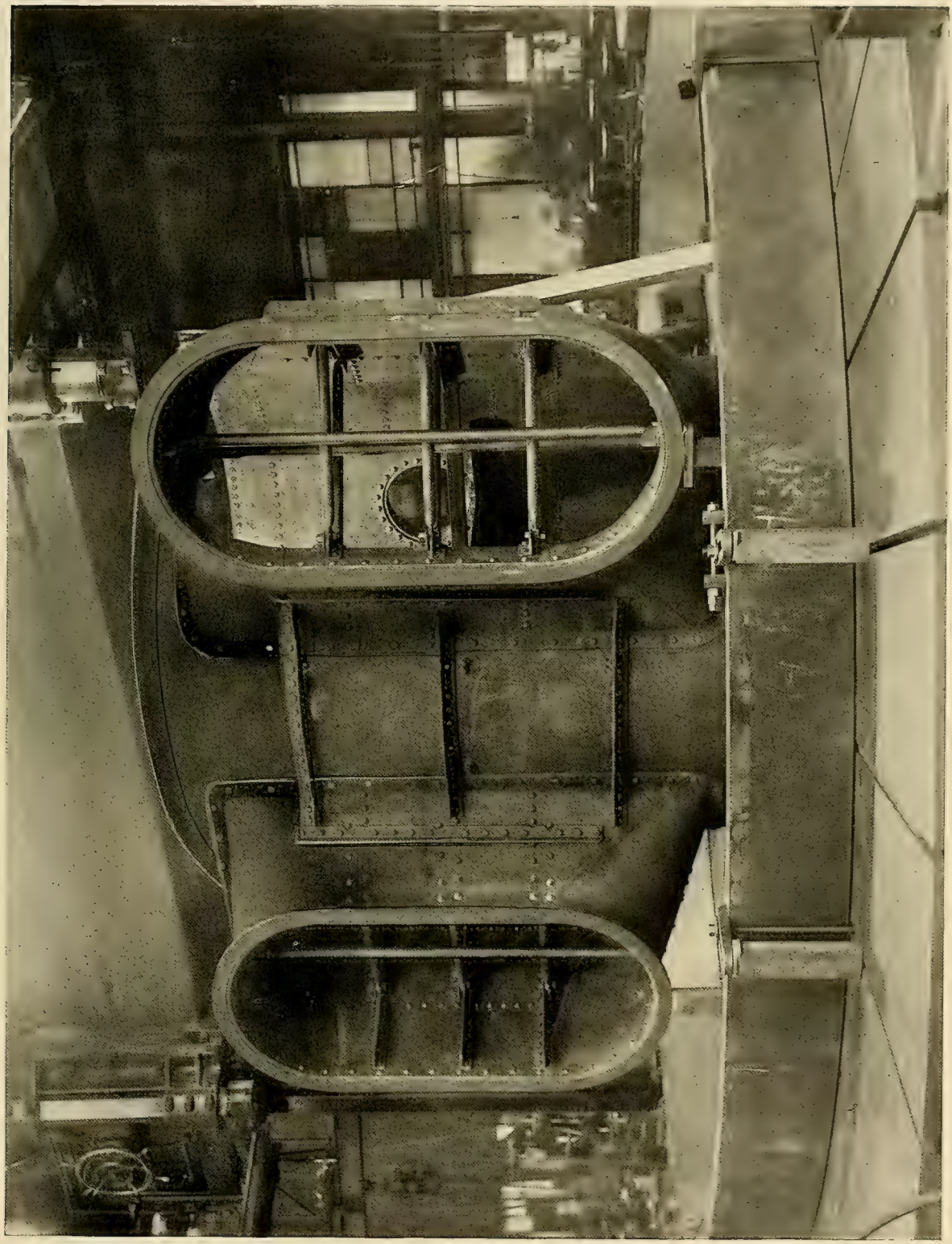
WM J. ROGERS, LITHO WASH., D C

To illustrate paper on "Surface Condensers," by Luther D. Lovekin, Esq., Member.



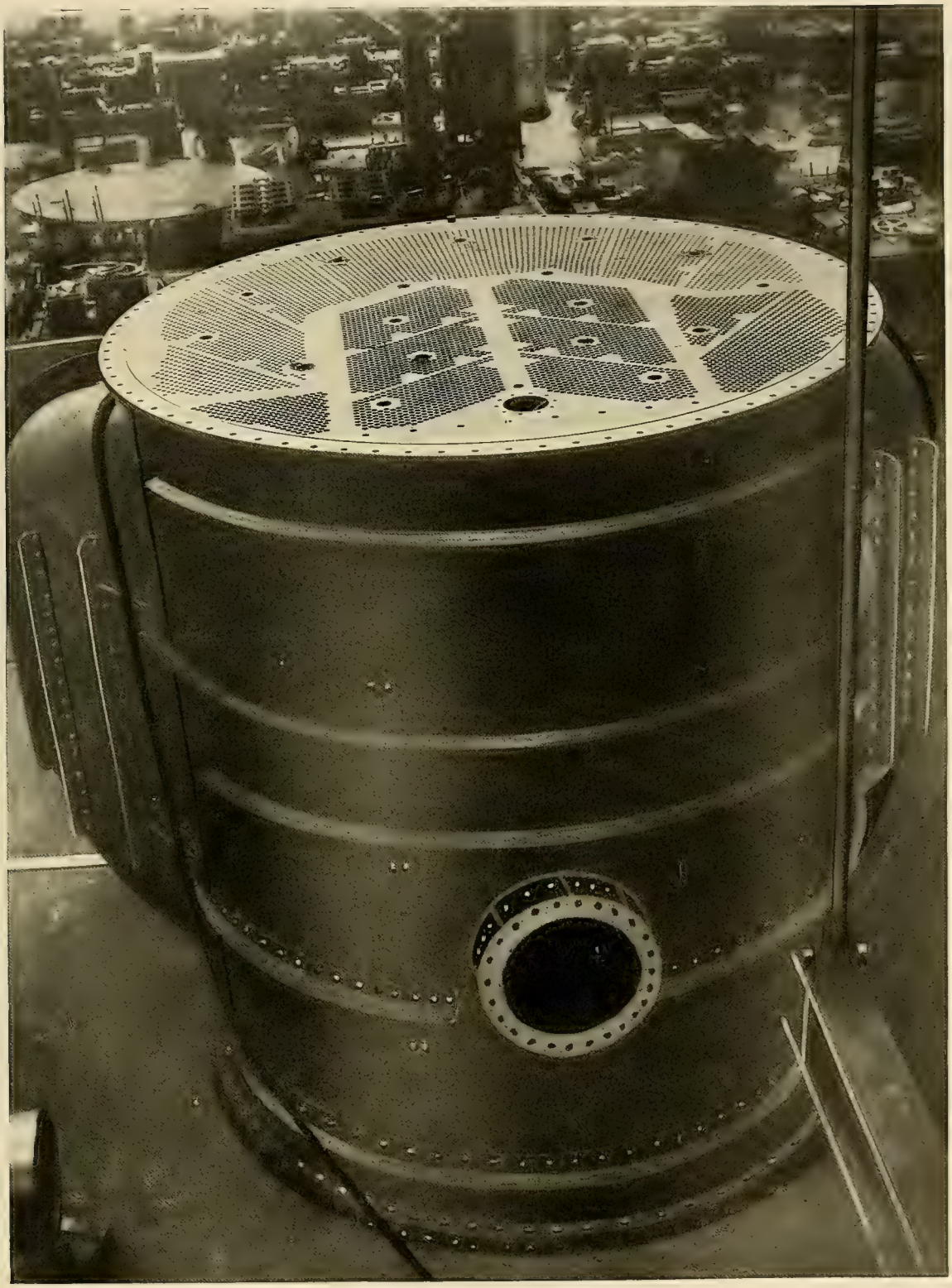
MAIN CONDENSER OF THE U. S. CANTIGNY AND CLASS.

To illustrate paper on "Surface Condensers,"
by Luther D. Lovekin, Esq., Member.



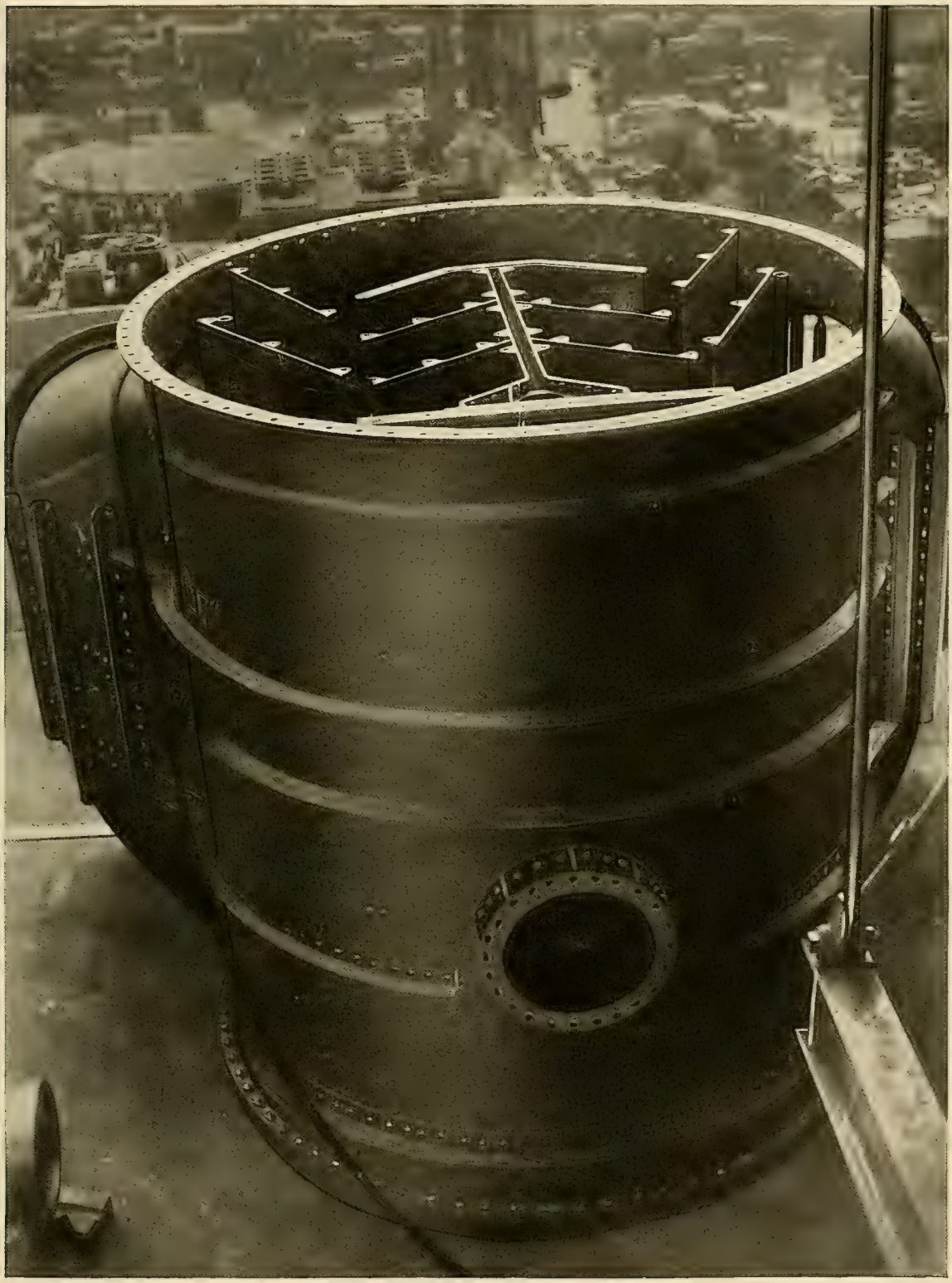
Main Condenser of the U. S. Battleship Tennessee and Class.

To illustrate paper on "Surface Condensers,"
by Luther D. Lovekin, Esq., Member.



Main Condenser of the U. S. Battleship Tennessee and Class.

To illustrate paper on "Surface Condensers,"
by Luther D. Lovekin, Esq., Member.



Main Condenser of the U. S. Battleship Tennessee and Class.

AMERICAN INTERNATIONAL SHIPBUILDING CORPORATION.

MAIN ENGINE ROOM LOG. SHEET "A"

FROM THREE HOUR READINGS TAKEN FROM THREE HOUR FULL SPEED NATURAL DRAFT RIVER TRIAL.

TEST FOR EFFICIENCY. — MAY 20 1920

U.S.A.T. CANTIGNY. — FROM TWO & ONE HALF " " " " " " " " INDUCED " " " — HULL No 670

DRAFT- FORD 17ft 0", AFT 21ft 9" MEAN 19ft 4½" —— PROPELLER DIA. 18ft 2⅝", PITCH 17.086ft"

TIME	STEAM PRESSURES: DRUM: FORD BOILER	DRUM: AFT BOILERS	DRUM: STAND-ARD.	THROTTLE: ON MAIN BOARD	THROTTLE: STAND-ARD.	AHEAD	ASTERN	VACUUM	LOW PRESS RECEIVER: AHEAD	LOW PRESS RECEIVER: ASTERN	AUX'Y STEAM	TEMPERATURES: THROT-TLE	FORD SUPERHEATER: IN	FORD SUPERHEATER: OUT	AFTER SUPERHEATER: IN	AFTER SUPERHEATER: OUT	AIR	ENG' ROOM	HOT WELL	SEA	FEED WATER HEATER. TEMPERATURES: WATER IN	WATER OUT	MAIN FEED PRESS	DOUBLE STROKE BOILER FEED PUMP	REVOLUTIONS: COUNTER	REVS PER ½ HOUR	AVERG' R.P.M.	ELECT TACHOMETR	VACUUM: SPRING GAUGE	MERCY COLUMN	GENERATING SET. PORT: AMPS.	PORT: VOLTS	PORT: R.P.M.	STARBOARD: AMPS.	STARBOARD: VOLTS	STARBOARD: R.P.M.
P.M.																																				
12.49	205	200	-	199	-	184	29"	5	26"	128	500	390	512	385	490	59	92	86	59.5	139	218	200	15	83597	-	-	92	28.80	28.95	102	120	400	100	119	400
1.19	206	207	-	201	-	185	29"	6	26"	130	499	390	513	386	460	59	91	84	59	138	217	210	16	86573	2976	99.20	91	28.90	29.00	98	120	400	102	120	400
1.49	200	197	205	192	187	176	29"	5	26"	130	488	388	502	385	440	59	90	85	58.5	140	228	220	16	89465	2892	96.40	92	28.90	29.00	96	118	400	101	120	405
2.19	201	200	198	193	188	177	29"	5	26"	128	500	388	521	385	512	59	90	86	58.5	142	237	220	16	92409	2944	98.13	92	28.80	28.95	97	119	400	103	121	410
2.49	205	204	204	200	195	183	29"	6	26"	130	494	380	518	390	505	59	97	85	58	143	234	215	14	95425	3016	100.20	100	28.25	28.55	98	119	405	102	119	410
3.19	203	202	204	197	195	180	29"	5.5	26"	130	495	388	510	388	510	59	92	84	57.5	144	224	215	15	98462	3037	101.23	98	28.90	28.95	98	118	405	101	120	405
3.49	206	205	204	200	195	183	29"	6	26"	130	494	388	516	398	505	59	101	84	57.5	143	227	215	16	101500	3038	101.27	101	28.90	28.97	96	119	405	102	118	408
																								DIFF.											
AVG.	203.7	202.1	203	197.4	192	181.1	29"	6.4	26"	129.4	482.9	387.4	514	386.7	488.8	59	93.3	84.9	58.3	141.6	226.4	215	15.4	17903	2973.8	99.41	95.1	28.78	28.91	97.9	119	402.1	101.4	119.6	405.4
CORR.						-1.0	-0.5	+5.0																												
CORR.																																				
AVG.				196.4	191.5	186.1								INDUCED DRAFT.																						
4.19	204	203	203	199	194	182	29"	5.5	26"	130	512	390	540	385	525	62	99	84	57	143	230	215	16	104544	-	-	100	28.90	28.98	98	118	405	101	120	405
4.49	207	206	205	200	196	184	29"	6	26"	130	496	390	530	390	495	63	100	87	57.5	145	232	220	16	107513	2969	98.97	-	28.90	28.90	99	119	410	102	120	415
5.19	186	185	178	172	167	156	29"	2	26"	126	504	380	520	380	510	63	103	84	58	144	230	220	16.5	110430	2917	97.23	97.0	29.00	29.07	99	120	410	100	118	405
5.49	200	200	205	199	194	183	29"	5.5	26"	130	508	390	530	385	505	62	100	86	57.5	142	234	220	16	113327	2897	96.57	100.5	28.95	28.92	99	120	410	101	120	400
6.19	201	201	199	193	187	177	29"	4.8	26"	130	516	385	520	384	518	62	102	87	60	144	227	215	16	116283	2956	98.53	98.0	28.95	28.97	98	120	410	101	121	410
6.49	205	206	201	196	191	180	29"	5.1	26"	130	514	380	522	385	516	60	102	83	60	144	224	220	16	119093	2810	93.67	93.0	28.95	28.89	99	120	405	102	120	410
																								DIFF.											
AVG.	200.5	200.2	198.5	193.2	187.3	177	29"	4.8	26"	128	508.3	385.8	527	384.8	511.5	62	101	85.2	58.7	143.7	229.5	218.3	16.1	14549	2909.4	96.99	99.7	28.92	28.92	98.7	119.5	408.3	101.2	119.8	407.5
CORR.				-1.0	-0.5	+5.0																														
CORR.																																				
AVG.				192.2	186.8	182.0																														

FOR LUBRICATING OIL DATA, & TURBINE BEARING TEMPERATURES, SEE SHEET "B."

FOR MAIN CONDENSER DATA, SEE SHEET "C."

FOR FIRE ROOM LOG SHEET SEE SHEET "D."

NATURAL DRAFT UNTILL 4^{co} P.M. INDUCED DRAFT THEREAFTER.

SUPERHEAT AT OUTLET OF FORD SUPERHEATER WAS 126.6° FOR NATURAL DRAFT & 141.2° FOR INDUCED DRAFT.

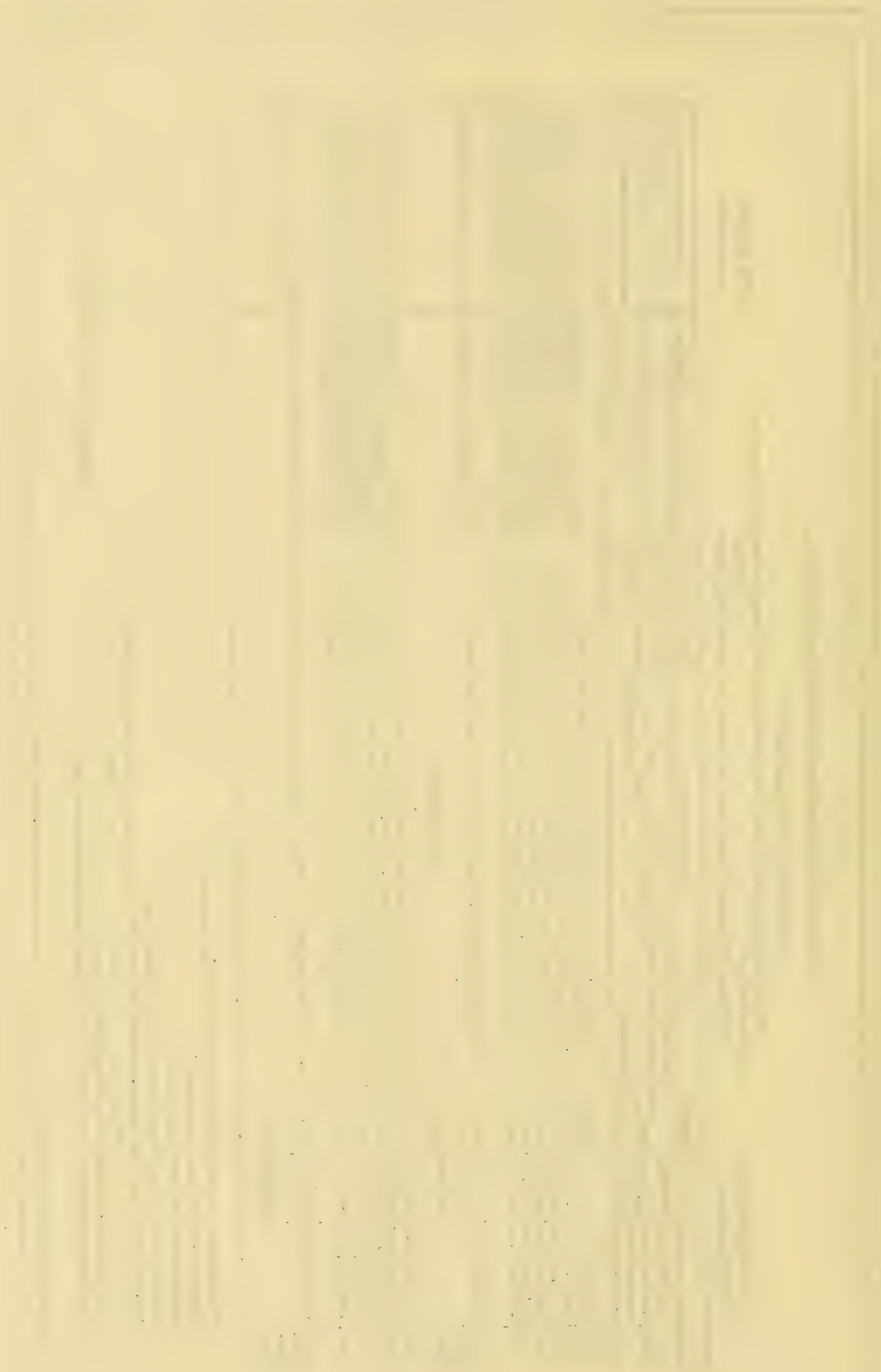
" " " " AFTER " " 102.2° " " " 126.7° " " "

FUEL OIL METERS AVERAGED 14.40% HIGH DURING ENTIRE TRIAL.

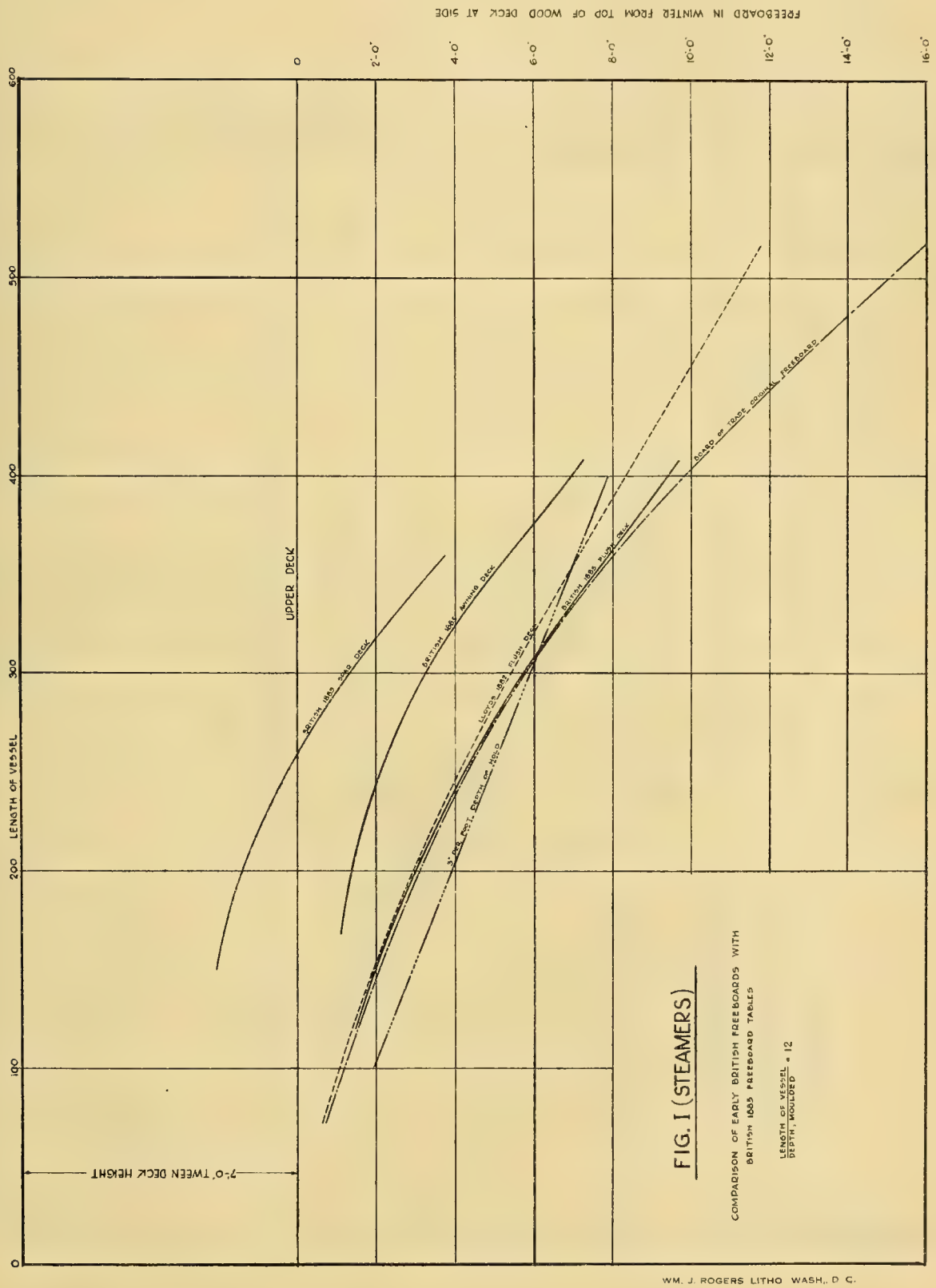
ALL MACHINERY WORKING VERY WELL. — FOR WEATHER CONDITIONS SEE DECK LOG.

SIGNED BY J.A. Wilson

WM. J. ROGERS THO. NASH



To illustrate paper on "Rules and Regulations for Freeboard," by David Arnott, Esq., Member.



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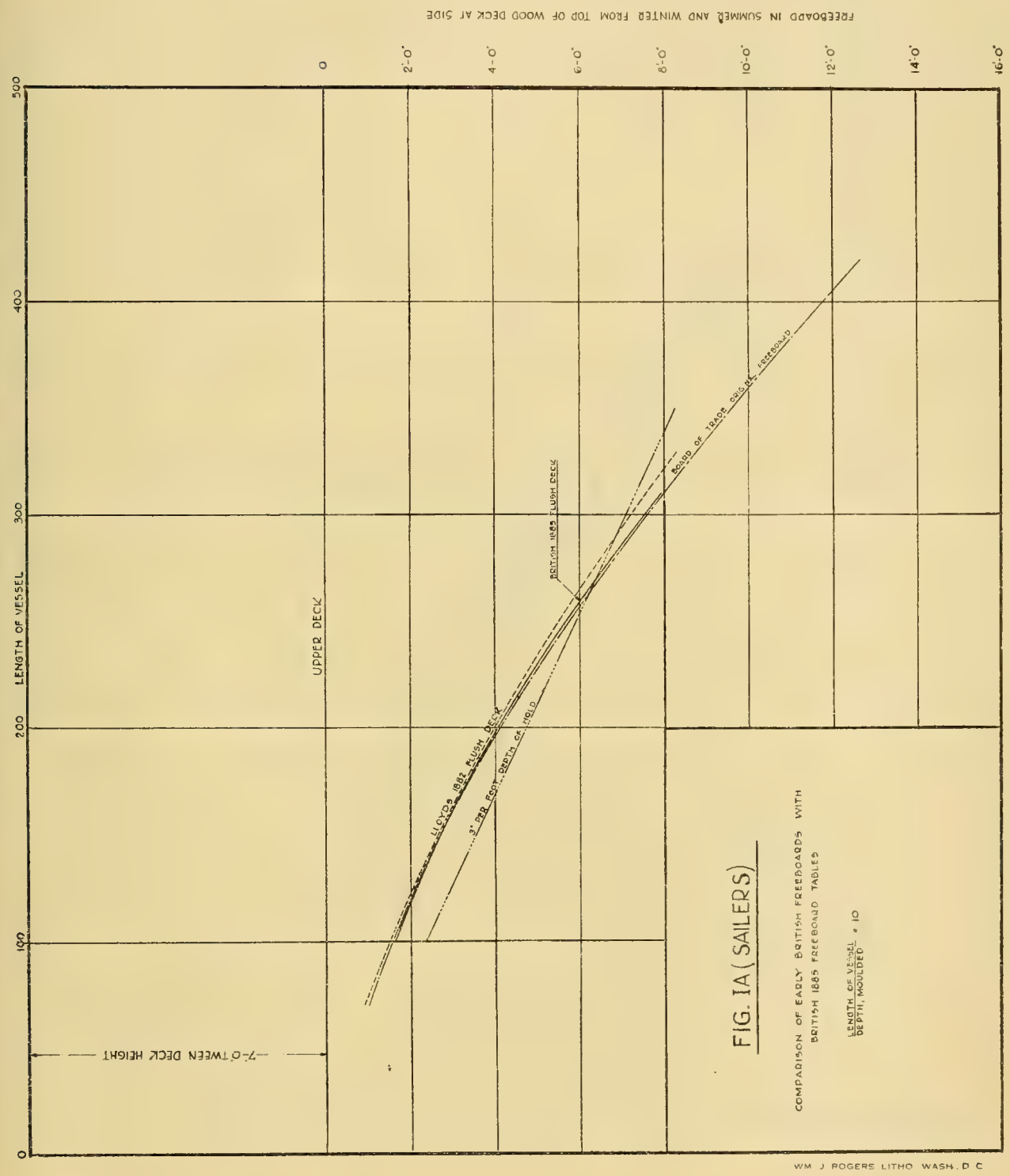

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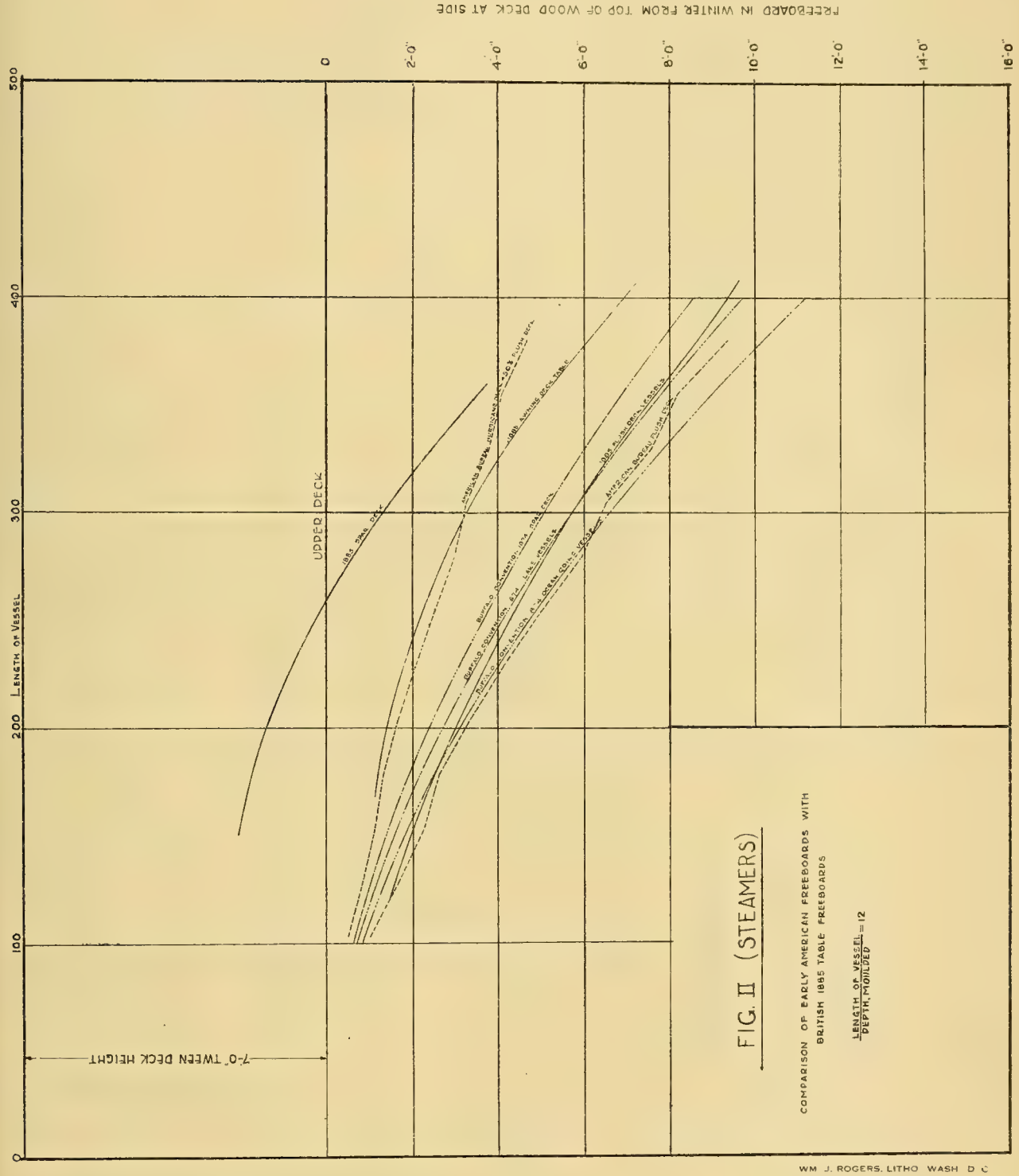

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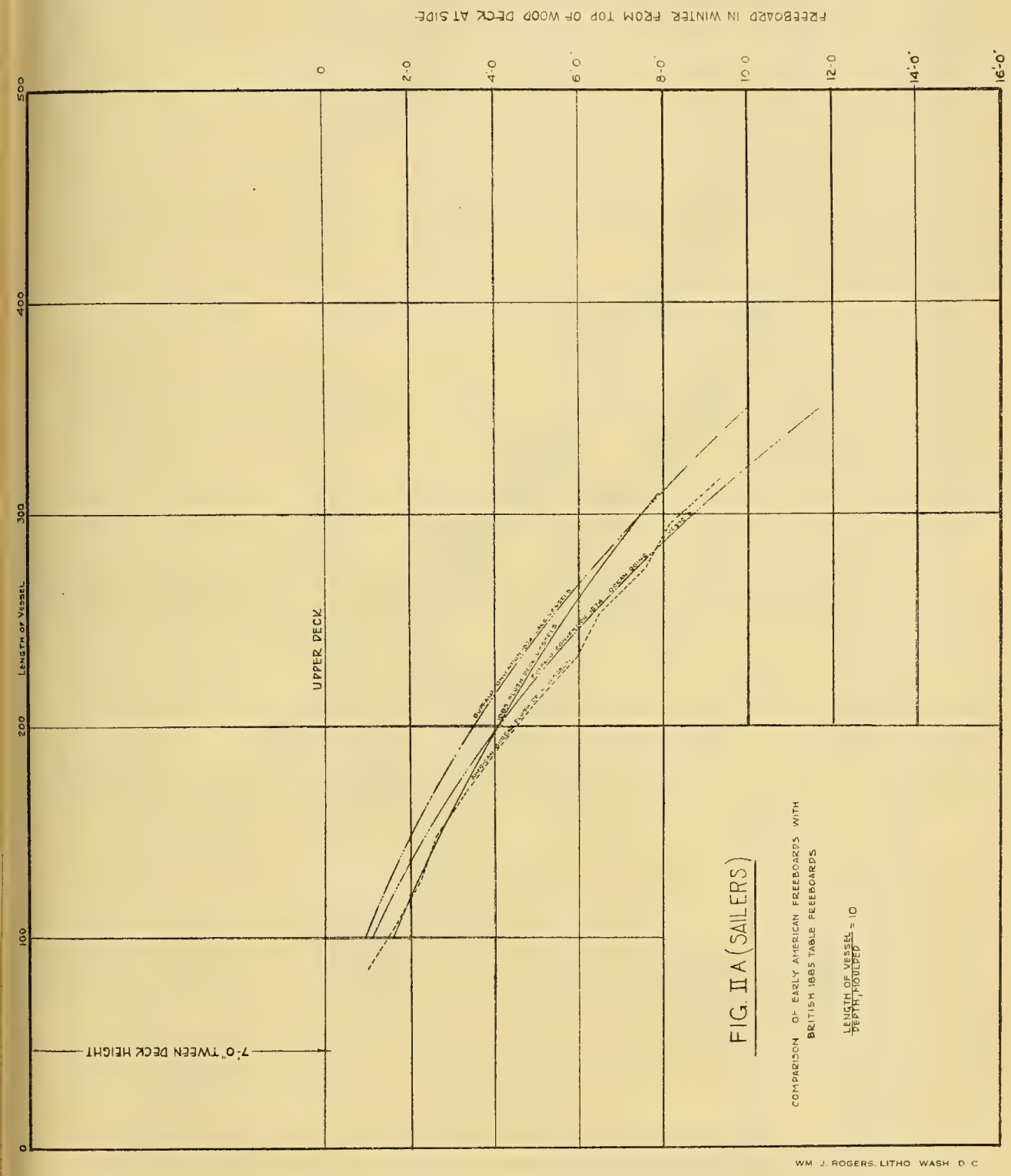

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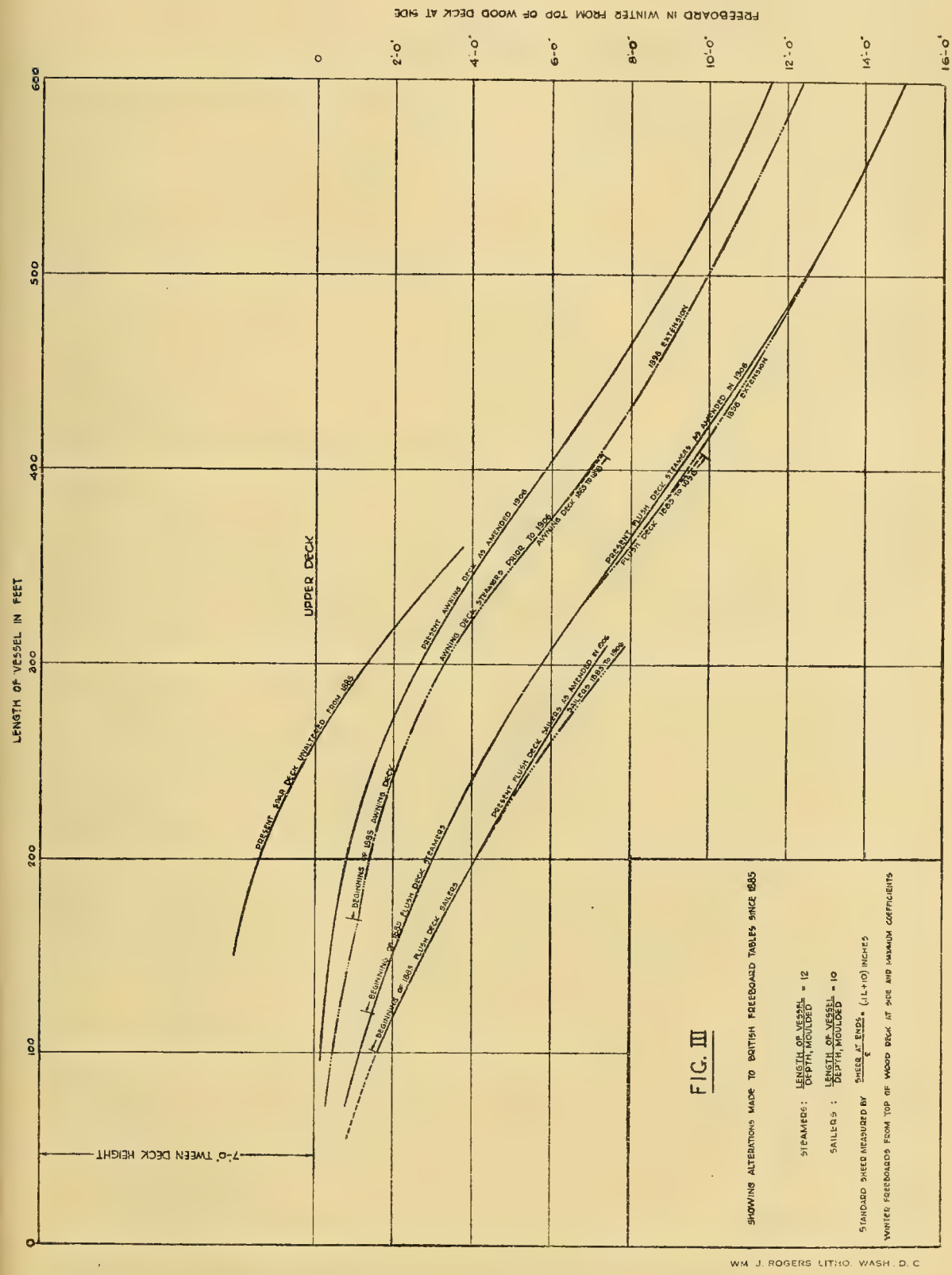
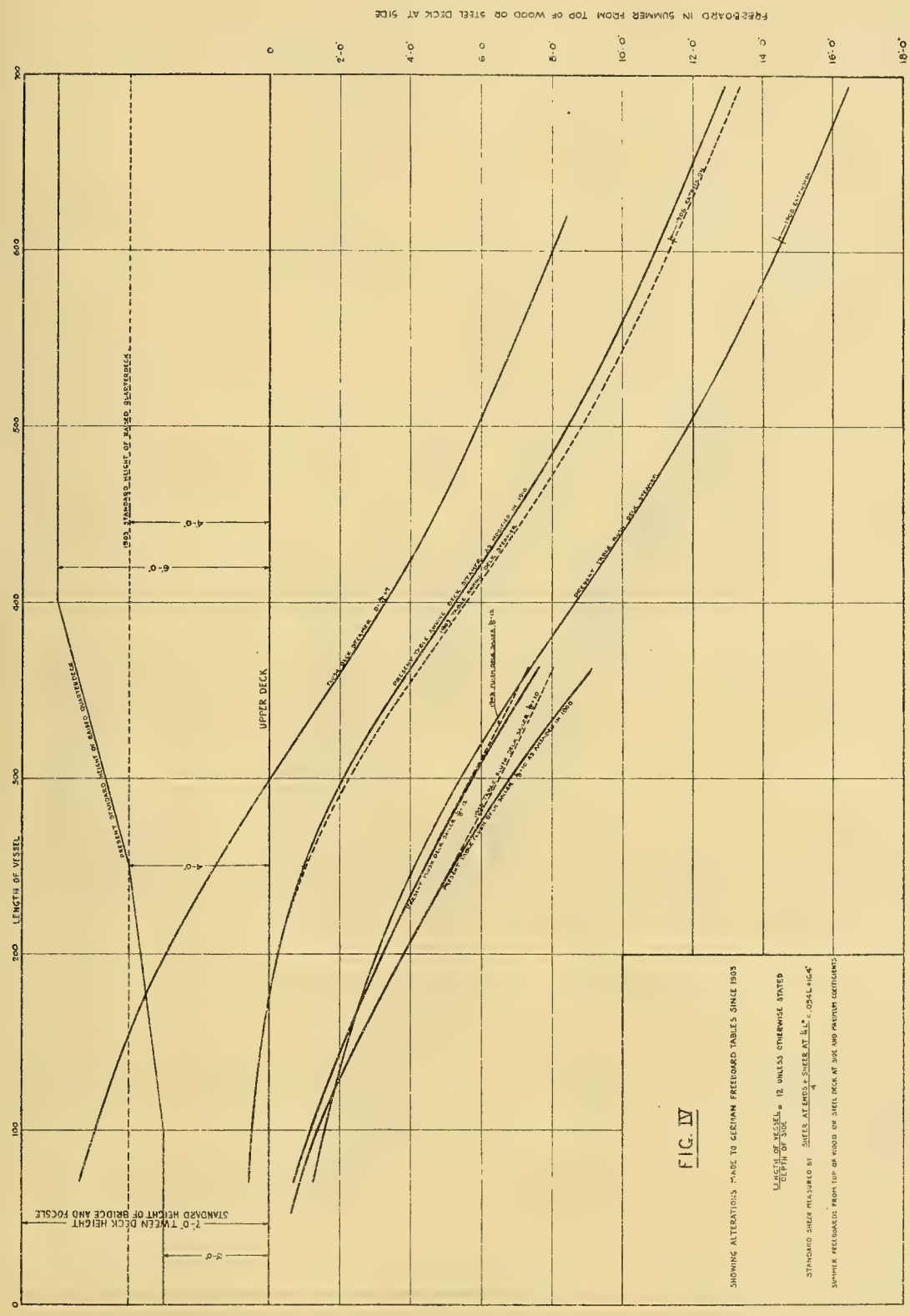


WM. J. ROGERS LITHO. WASH., D. C.

To illustrate paper on "Rules and Regulations for Freeboard," by David Arnott, Esq., Member.



WM. J. ROGERS, LITHO., WASH., D. C.

To illustrate paper on "Rules and Regulations for Freeboard," by David Arnott, Esq., Member.

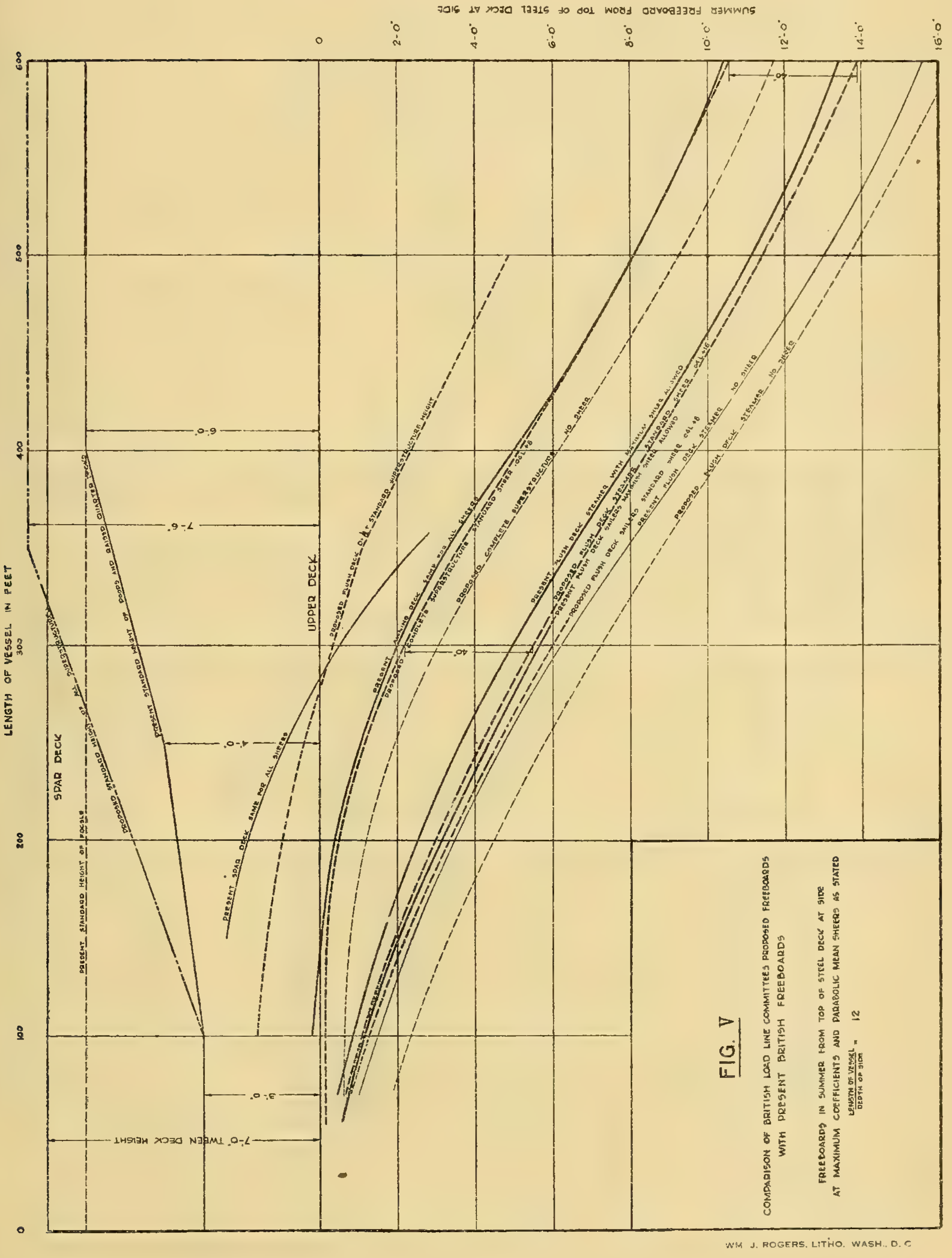
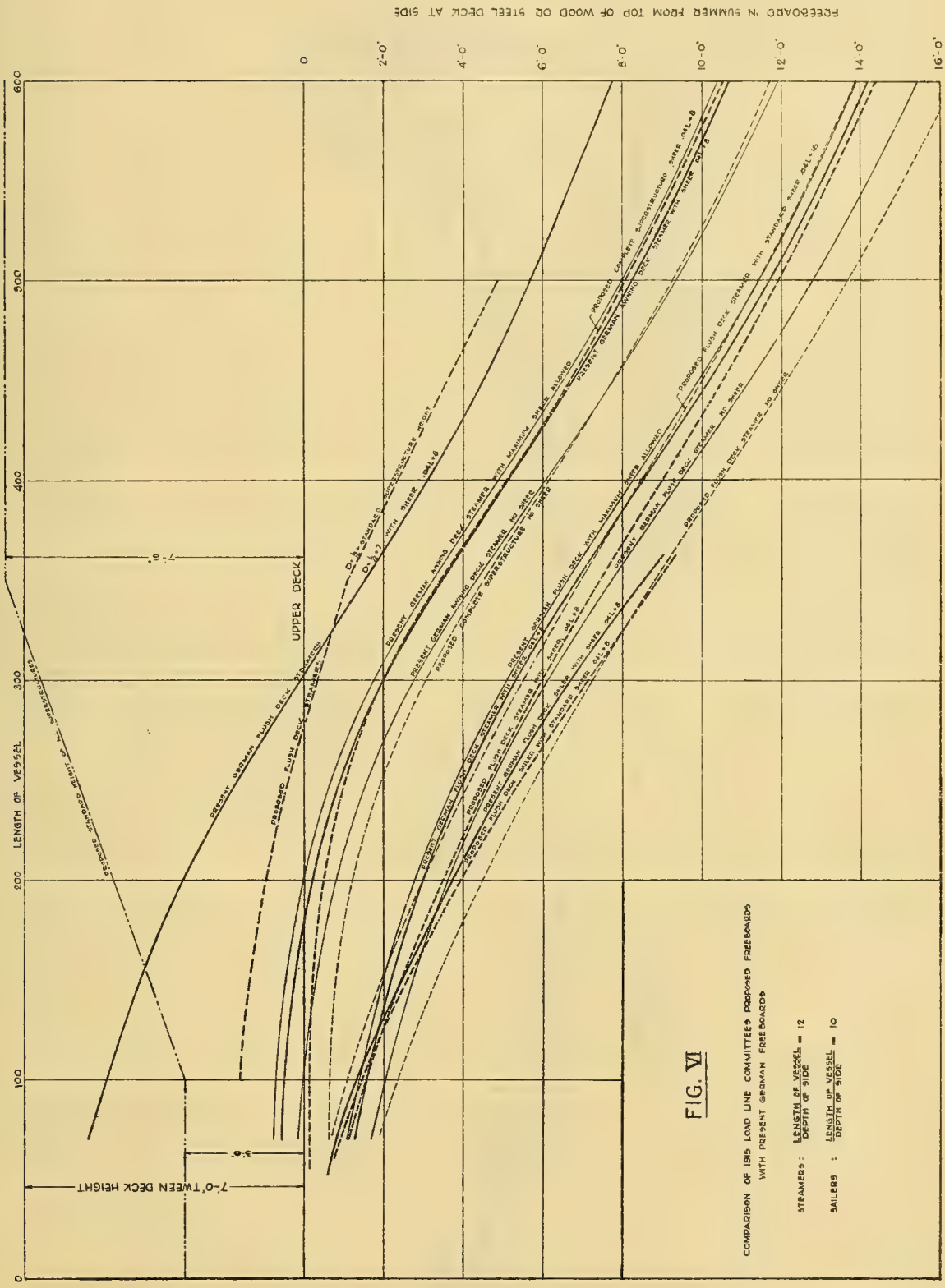


FIG. V

WM. J. ROGERS, LITHO. WASH., D. C.

To illustrate paper on "Rules and Regulations for Freeboard."
by David Arnott, Esq., Member.



WM. J. ROGERS, LITHO., WASH., D. C

To illustrate paper on "Rules and Regulations for Freeboard," by David Arnott, Esq., Member.

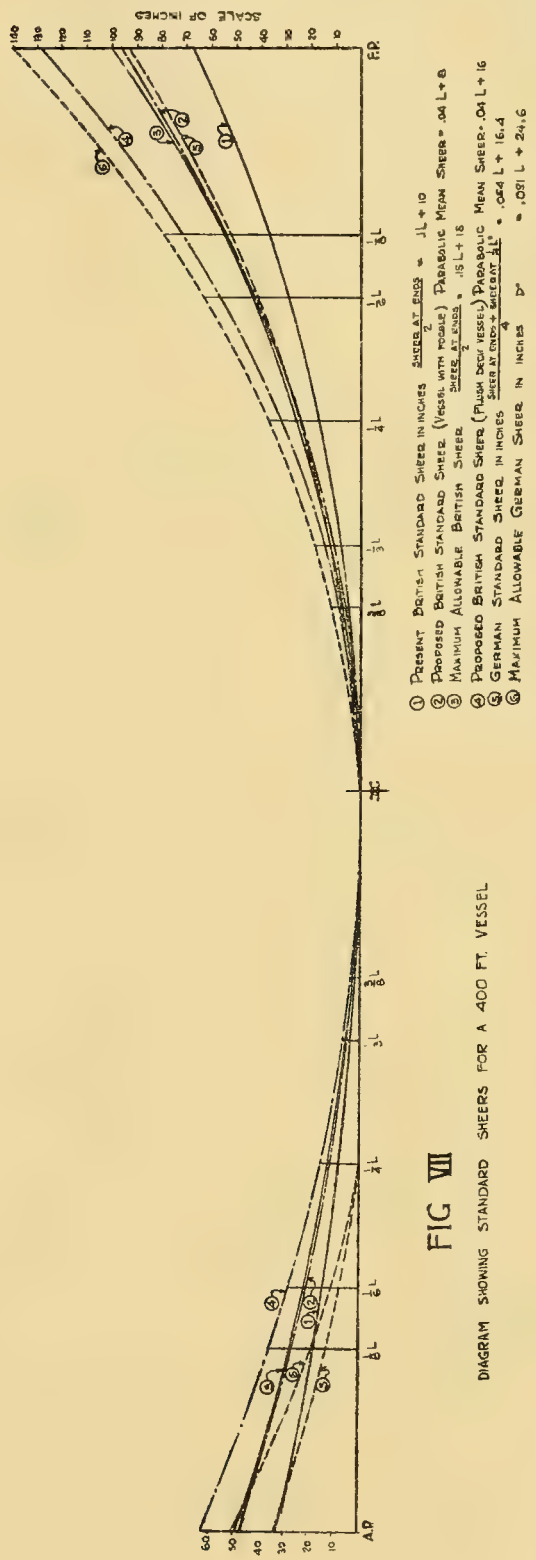


FIG VII

DIAGRAM SHOWING STANDARD SHEERS FOR A 400 FT. VESSEL

ROGERS, LITHO., WASH., D. C.

To illustrate paper on "Rules and Regulations for Freeboard," by David Arnott, Esq., Member.

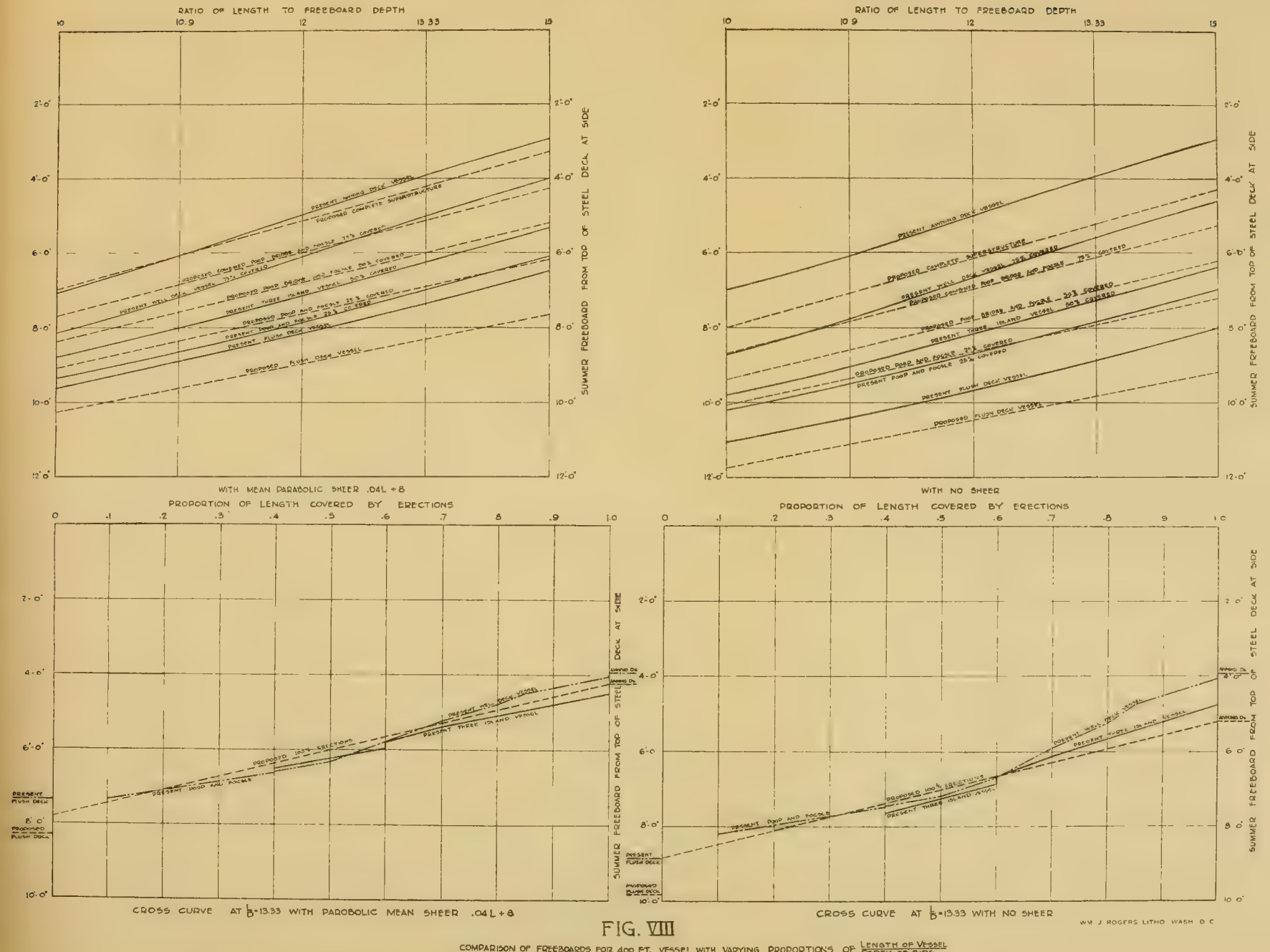
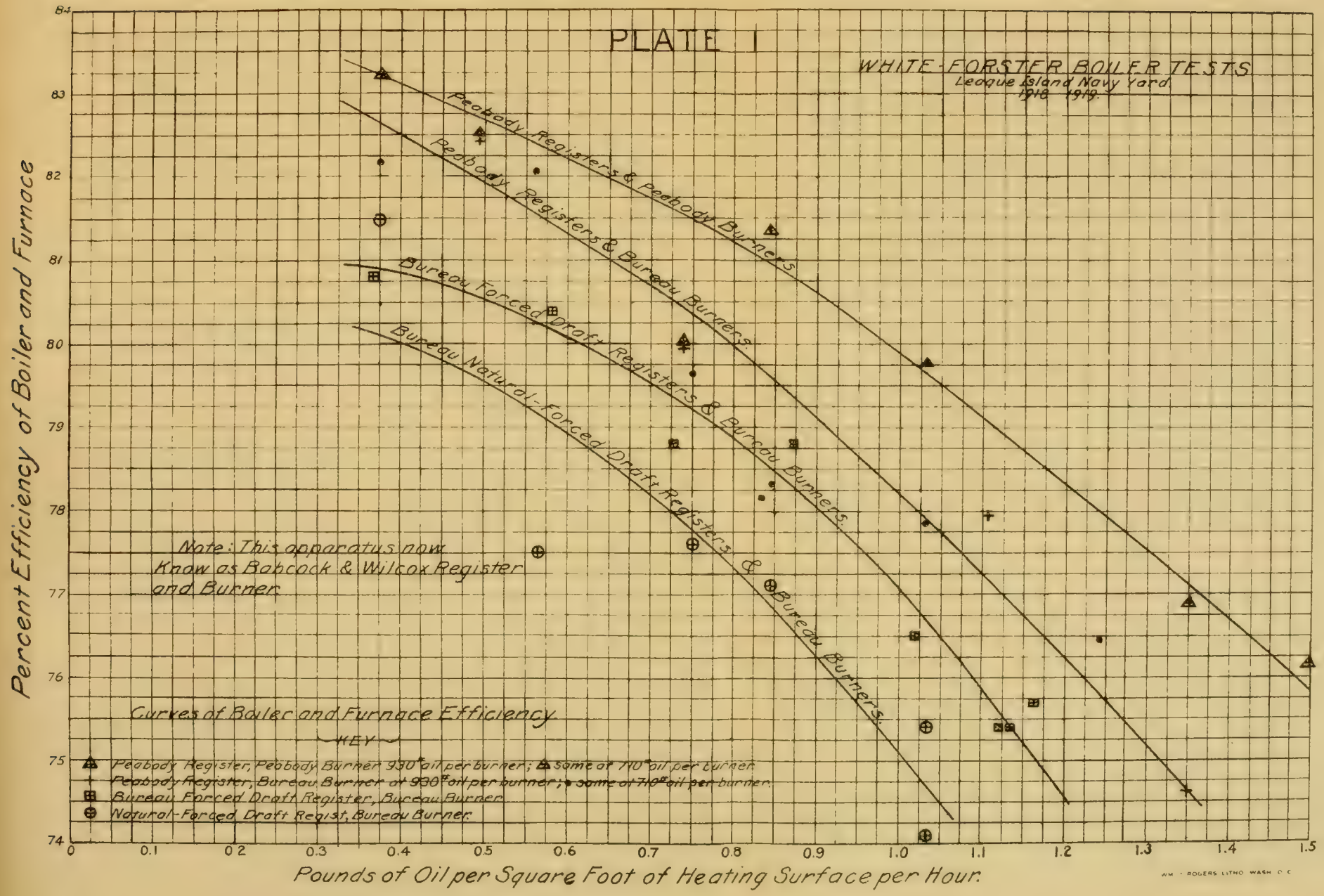


FIG. VIII

COMPARISON OF FREEBOARDS FOR 400 FT. VESSEL WITH VARYING PROPORTIONS OF $\frac{\text{LENGTH OF VESSEL}}{\text{DEPTH OF SIDE}}$

To illustrate paper on "Recent Advance in Oil Burning," by Ernest H. Peabody, Esq., Member.



To illustrate paper on "Recent Advance in Oil Burning," by Ernest H. Peabody, Esq., Member.

FUEL OIL TESTING PLANT

EVAPORATIVE TEST ON WHITE FORSTER BOILER USING PEABODY AIR REGISTERS & BUREAU STANDARD OIL ATOMIZERS.

Water heating surface 7565 Sq. ft.
Superheating surface 753 Sq. ft.
Total heating surface 8318 Sq. ft.
Furnace Volume 751 Cu. ft.

Number of tubes 4030.
Size of tubes 1" OD
Diameter of steam drum 54"
Working pressure 300 lbs. per sq inch gauge

1 Test Number	32-2	32-3	32-4	32-5	32-6	32-7	32-8	32-9	32-10	32-11
2 Date	April 12, 1919	April 14, 1919	April 14, 1919	April 16, 1919	April 16, 1919	April 16, 1919	May 14, 1919	May 17, 1919	May 17, 1919	May 16, 1919
3 Barometer - Inches of mercury	30.06	30.02	29.98	29.92	29.81	29.72	29.91	29.81	29.79	29.99
4 Duration of test	3 hrs.	3 hrs.	3 hrs.	3½ hrs	3 hrs	1½ hrs	3 hrs	3 hrs	2½ hrs	4 hrs
5a Number of burners used	4	6	8	9	11	11	4	6	9	11
6b Size of atomizers	1/16 x 1/16 - 7/64	1/16" x 1/16" - 7/64	1/16 x 1/16 - 7/64	1/16 x 1/16 - 7/64	1/16 x 1/16 - 7/64	1/16 x 1/16 - 7/64	5/64 x 1/16 - 1/8	5/64 x 1/16 - 1/8	5/64 x 1/16 - 1/8	5/64 x 1/16 - 1/8
c Arrangement of burners used	8-9-10-11	4-6 8-9-10-11	3-4-6-7 8-9-10-11	3-4-5-6-7 8-9-10-11	All	All	8-9-10-11	4-6 8-9-10-11	4 - 1-2 5 - 6 8-9-10-11	All.
6a Type of air registers used	Peabody	Peabody	Peabody	Peabody	Peabody	Peabody	Peabody	Peabody	Peabody	Peabody
b Aggregate opening of air registers Sq Inches	481.6	722.4	963.2	1083.6	1324.4	1324.4	481.6	722.4	10836	1724.4
STEAM OBSERVATIONS										
7 Steam pressure on drum - Lbs per Sq inch gauge	276.0	276.0	276.0	276.0	276.0	275.9	296.0	296.4	295.8	296.0
8 Steam pressure below boiler stop Lbs per sq inch gauge	275.8	271.4	266.1	262.8	253.2	242.1	286.8	279.7	267.5	255.1
9 Superheater temperature °F	446.8	460.3	472.2	473.5	470.8	467.2	460.2	476.5	4898	494.9
10 Calorimeter temperature (lower thermometer) °F	311.9	312.0	310.9	310.3	297.3	292.7	311.9	311.0	310.0	306.9
11 Calorimeter inlet steampressure) lbs per sq inch gauge	275.5	273.5	272.0	270.5	267.5	263.9	294.0	292.9	286.3	273.5
12 Feedwater temperature at boiler °F	198.1	198.0	198.9	1984	198.3	198.8	194.2	196.2	197.0	195.6
13 Water actually evaporated Lbs. per hour	41735.	61949	79659	88613.	107857.	127560	54887.	79018.	114831.	134239.
FUEL OBSERVATIONS										
14 Oil burned - Lbs. per hour	28397	4257.0	56763	6394.3	7821.7	9410.7	3720.7	5580.7	8374.4	102280
15 Oil pressure at burners - Lbs. per sq. inch gauge	1746	176.1	170.1	160.9	161.6	2299	148.6	148.7	148.4	1488
16 Oil temperature at burners - °F	119.3	120.9	121.3	120.0	120.3	120.4	119.7	120.1	119.8	120.6
AIR OBSERVATIONS										
17 Dry bulb temperature - °F	798	81.9	80.5	67.0	67.3	69.7	93.4	90.8	79.8	77.3
18 Wet bulb temperature °F	60.1	61.1	60.7	58.9	59.5	63.3	70.7	75.2	67.7	63.4
19 Fireroom temperature °F	92.3	96.7	96.2	81.9	82.2	84.8	101.7	99.6	90.5	85.1
20a Stack temperature - °F	472.8	522.0	571.4	5976	652.0	681.4	514.	581.	673.	723.
b Furnace temperature (by optical pyrometer) - °F	2770.	2950	2980.	2990	3025.	3055	2870.	3090.	3010	3060
21 Fireroom presure - Inches of water	259	360	4.67	5.19	6.38	7.83	4.353	5976	8.085	9.543
22 Air pressure in furnace Inches of water	.12	.39	.83	1.18	1.95	2.52	.294	.852	2.138	3.293
23 Air pressure at base of stock Inches of water	- 12	- .12	- .13	- .10	- 10	- .10	- .138	- .147	- .098	- .071
24 Smoke by Ringelmann scale	0 - ¼	0	¼ - 1	¼ - ¾	½ - ¾	½ - 1	0 - ¼	0 - ¼	0 - ¼	0 - ¼
FLUE GAS ANALYSIS										
24½ Sampling tube numbers	6 / 1-2• / 3	6 / 1-2• / 3	6 / 1-2• / 3	6 / 1-2• / 3	6 / 1-2• / 3	6 / 1-2• / 3	11-12 18-19 / 3 / 2	11-12 18-19 / 6-5 / 2-4	11-12 18-19• / 6-15 / 2-4	11-12 18-19• / 6-15 / 2
25 Carbon dioxide - %	12.67 / 12.53 / 12.23	11.46 / 12.25 / 11.63	12.53 / 11.80 / 12.16	14.55 / 11.50 / 10.86	14.46 / 11.43 / 9.60	13.40 / 11.85 / 9.96	13.19 / 11.10 / 14.00	12.70 / 14.05 / 11.25	13.10 / 13.57 / 12.85	13.51 / 13.39 / 13.51
26 Oxygen %	3.15 / 3.60 / 3.70	4.57 / 3.89 / 4.10	3.03 / 4.67 / 4.10	.95 / 4.95 / 5.30	1.117 / 5.38 / 7.26	1.70 / 4.15 / 6.70	3.39 / 5.20 / 1.60	3.71 / 1.75 / 4.82	2.74 / 2.55 / 3.47	2.42 / 2.40 / 1.95
27 Carbon monoxide %	.00 / .00 / .00	.00 / .00 / .00	.00 / .00 / .00	.05 / .00 / .00	.03 / .00 / .00	.00 / .00 / .00	.00 / .00 / .00	.00 / .00 / .01	.12 / .00 / .08	.24 / .40 / .00
28 Nitrogen by difference %	84.18 / 83.87 / 84.07	83.97 / 83.86 / 84.27	84.44 / 83.53 / 83.74	84.45 / 83.55 / 83.84	84.34 / 83.19 / 83.14	84.90 / 84.00 / 83.35	83.42 / 83.70 / 84.40	83.59 / 84.20 / 83.92	84.04 / 83.88 / 83.60	83.83 / 83.87 / 84.54
ULTIMATE ANALYSIS & PHYSICAL TESTS OF FUEL										
29 Carbon %	84.89	84.89	84.89	84.90	84.90	84.90	85.42	85.36	85.36	85.50
30 Hydrogen %	12.01	12.01	12.01	12.31	12.31	12.31	11.73	11.76	11.76	11.79
31 Sulphur %	.95	.95	.95	.86	.86	.86	1.16	.98	.98	1.18
32 Nitrogen %	.29	.29	.29	.29	.29	.29	.24	.25	.25	.18
33 Oxygen and undetermined %	1.86	1.86	1.86	1.64	1.64	1.64	1.45	1.65	1.65	1.35
34 Gravity - Baume	22.95°	22.95°	22.95%	23.00°	23.00°	23.00°	24.85	25.5	25.5	25.8
35 Gravity Specific	.9162	.9162	.9162	.9137	.9137	.9137	.9051	.9014	.9014	.8995
36 Flash point - °F	160	160	160	160	160	160	210	255	255	245
37a Viscosity temperature °F	120	120	120	120	120	120	104.	104.	104.	104.
b Viscosity - Engler	3.24	3.24	3.24	3.04	3.04	3.04	6.54	4.07	4.07	4.00
38 Water & sediment %	.5%	.5%	.5%	.4%	.4%	.4%	.4%	.2%	.2%	.2%
39 Calorific value B.T.U. per lb	18998	18998	18998.	19102.	19102	19102.	19216	19166	19166.	19256.
STEAM RESULTS										
40 Superheat - °F	32.2	47.1	60.7	63.0	63.5	63.8	42.3	60.7	77.8	86.9
41 Quality of steam leaving drum %	99.19	99.21	99.15	99.13	98.41	98.18	99.07	99.04	99.01	98.85
42 Factor of evaporation	1.0944	1.1041	1.1109	1.1124	1.1119	1.1105	1.1064	1.1149	1.1229	1.1281
43 Actual evaporation per lb. of fuel - lbs	14.697	14.552	14.034	13.858	13.790	13.555	14.752	14.159	13.712	13.125
44 Equivalent evaporation per lb. of fuel lbs	16.084	16.067	15.590	15.416	15.333	15.053	16.322	15.786	15.397	14.806
45 Actual evaporation per sq. ft. of water H.S. lbs per hr.	5.517	8.189	10.530	11.714	14.257	16.862	7.255	10.445	15.179	17.745
46 Equivalent evaporation per sq. ft. water H.S. lbs per hour	6.038	9.042	11.698	13.031	15.852	18.725	8.027	11.645	17.044	20.018
47 % of total heat transfer taking place in superheaters	2.92%	3.75%	4.47%	4.59%	5.10%	5.22%	3.61%	4.57%	5.36%	5.80%
OIL RESULTS										
48 Oil burned per sq. ft. of water H.S. lbs. per hour	.375	.563	.750	.845	1.034	1.244	.492	.738	1.107	1.352
49 Oil burned per cu. ft. of furnace volume lbs per hr.	3.781	5.668	7.558	8.514	10.415	12.531	4.954	7.431	11.151	13.619
50 Oil burned per burner per hour lbs	709.9	709.5	709.5	710.5	711.1	855.5	930.2	930.1	930.5	929.8
AIR RESULTS										
51 Relative Humidity %	29.2	27.5	31.9	60.3	62.8	70.4	31.1	48.1	49.5	45.1
52 Dew Point - °F	44.4	45.4	45.6	53.0	54.7	59.8	59.0	68.6	59.0	54.5
53a Moisture - lbs. per cu ft.	.00048	.00050	.00050	.00065	.00069	.00092	.00080	.00110	.00080	.00069
b Vapor pressure (e) - Inches of Mercury	.291	.303	.305	.402	.428	.513	.499	.698	.499	.424
54 Weight of dry air - lbs. per cu. ft.	.07197	.07145	.07158	.07320	.07307	.07255	.07022	.07006	.07201	.07367
55 Weight of air at atmospheric conditions - lbs per cu ft.	.07245	.07195	.07208	.07385	.07376	.07337	.07102	.07116	.07281	.07436
56 Air at atomospheric conditions theoretically required for combustion of one lb of fuel - cu. ft.	193.6	195.0	194.3	191.4	191.8	193.1	197.9	1981	192.7	189.1
RESULTS FROM ORSATT & ULTIMATE ANALYSIS										
57 Weight of dry gasses per lb. of fuel - lbs	17.02	17.39	18.03	18.48	18.60	17.96	16.32	16.90	16.25	15.68
58 Weight of dry air per lb. of fuel - lbs.	17.22	17.61	18.21	18.69	18.72	18.24	16.37	17.02	16.44	15.80
59 Excess air from Orsatt - %	19.7	21.6	27.3	29.4	32.9	23.3	18.4	20.5	14.3	12.4
60 Volume of air at atmospheric conditions used per lb. of fuel - Cu. ft	—	—	—	—	—	—	—	—	—	—
61 Volume of air at fireroom conditions used per lb. of fuel - Cu. ft	239.3	246.5	254.4	255.3	256.2	251.4	233.1	242.9	228.3	214.5
62 Velocity of air through vanes of air registers ft. per minute	3398.	3498.	3610	3628.	3644.	4301	4337	4518.	4249.	3989.
63 % excess air over theoretical amount required for complete combustion	23.6	26.4	30.9	33.4	33.6	30.2	17.8	22.6	18.5	11.3
HEAT BALANCE (B.T.U. / %)	B.T.U. / %	B.T.U. / %	B.T.U. / %	B.T.U. / %	B.T.U. / %	B.T.U. / %	B.T.U. / %	B.T.U. / %	B.T.U. / %	B.T.U. / %
64 Heat absorbed by boiler	15608 / 82.16	15591 / 82.07	15129 / 79.64	14960 / 78.32	14879. / 77.89	14607 / 76.47	15839. / 82.43	15319. / 79.93	14941. / 77.96	14368 / 74.62
65 Loss by imperfect combustion	0 / —	0 / —	0 / —	30 / .16	18 / .09	0 / —	0. / —	0. / —	79. / .41	152. / .79
66 Loss with dry chimney gases	1554 / 8.18	1775 / 9.34	2056 / 10.82	2287 / 11.97	2544 / 13.32	2572 / 13.47	1606. / 8.36	1946. / 10.15	2263. / 11.81	2389. / 12.40
67a Loss with moisture from hydrogen cu ft	1311 / 6.90	1331 / 7.01	1357 / 7.14	1420 / 7.43	1448 / 7.58	1460 / 7.64	1291. / 6.72	1329. / 6.93	1385 / 7.22	1419. / 7.37
b Loss with moisture in fuel	— / —	— / —	— / —	— / —	— / —	— / —	— / —	— / —	— / —	— / —
c Loss with moisture in air	21 / .11	25 / .13	28 / .15	40 / .21	47 / .25	58 / .30	36. / .19	60. / .31	50. / .26	44 / .23
68 Radiation and error	504 / 2.65	276 / 1.45	428 / 2.25	365 / 1.91	166 / .87	405 / 2.12	444. / 2.30	512. / 2.67	448. / 2.34	884 / 4.59
69 Total	18998 / 100.00	18998 / 100.00	18998 / 100.00	19102 / 100.00	19102 / 100.00	19102 / 100.00	19216. / 100.00	19166 / 100.00	19166. / 100.00	19256. / 100.00
70 Boiler and furnace efficiency %	82.16	82.07	79.64	78.32	77.89	76.47	82.43	79.93	77.96	74.62

Remarks - Readings used in calculations.

To illustrate paper on "Recent Advance in Oil Burning," by Ernest H. Peabody, Esq., Member.

FUEL OIL TESTING PLANT

EVAPORATIVE TEST ON WHITE FORSTER BOILER
USING PEABODY AIR REGISTERS
& PEABODY OIL ATOMIZERS

Water heating surface 7565 Sq. ft.
Superheating surface 753 Sq. ft.
Total heating surface 8318 Sq. ft.
Furnace volume 751 Cu. ft.

Number of tubes 4032
Size of tubes 1" O.D.
Diameter of steam drum 54"
Working pressure 300 Lbs. per sq. inch gauge

No.	Item								
1	Test Number	30-14	30-15	30-16	30-17	30-18	30-19	30-20	
2	Date	June 9 1919	June 9 1919	June 10 1919	June 10 1919	June 13 1919	June 13 1919	June 13 1919	
3	Barometer - Inches of mercury	30.29	30.26	30.32	30.32	30.40	30.32	30.26	
4	Duration of test	3 hrs.	3 hrs.	3 hrs.	2 hrs.	3 hrs.	2½ hrs.	2 hrs.	
5a	Number of burners used	4	6	11	11	4	9	11	
b	Size of atomizers	#104	#104	#104	#104	#103	#103	#103	
c	Arrangement of burners used	8-9-10-11	4—6 8-9-10-11	All	All	8-9-10-11	4-1-3-2-6 8-9-10-11	All	
6a	Type of air registers used	Peabody	Peabody	Peabody	Peabody	Peabody	Peabody	Peabody	
b	Aggregate opening of air registers - Sq. inches	481.6	722.4	1324.4	1324.4	481.6	1083.6	1324.4	
	STEAM OBSERVATIONS								
7	Steam pressure on drum - Lbs. per sq. inch, gauge	296.1	295.8	295.2	295.6	297.1	296.0	295.9	
8	Steam pressure below boiler stop - Lbs. per sq. inch gauge	292.1	285.5	254.2	245.5	284.9	273.3	272.9	
9	Superheater temperature - °F.	465.3	480.6	486.5	484.5	459.5	482.9	483.6	
10	Calorimeter temperature (lower thermometer) °F	315.5	314.7	302.4	296.7	315.3	—	—	
11	Calorimeter (inlet steam pressure) Lbs. per sq. inch gauge	294.1	292.3	283.2	281.6	296.1	—	—	
12	Feedwater temperature at boiler - °F.	197.0	196.9	196.9	198.0	195.1	194.7	197.2	
13	Water actually evaporated - Lbs. per hour	55323.	79632	139452.	153467	42523.	92203.	110663.	
	FUEL OBSERVATIONS								
14	Oil burned - Lbs. per hour	3726	5594	10231.	11355.	2839.	6370	78.0	
15	Oil pressure at burners - Lbs. per sq. inch gauge	197.2	197.6	197.3	245.9	169.4	170.6	169.3	
16	Oil temperature at burners - °F.	119.6	119.6	120.0	120.0	119.9	120.0	120.0	
	AIR OBSERVATIONS								
17	Dry bulb temperature - °F.	89.8	88.1	86.5	90.5	104.5	101.9	100.3	
18	Wet bulb temperature - °F.	75.9	73.0	74.5	75.8	77.6	71.7	70.4	
19	Fireroom temperature - °F	10[illegible]	99.3	97.8	109.1	115.1	115.5	114.6	
20a	Stack temperature - °F.	532.	614.	715.	729.	490.	617.	648.	
b	Furnace temperature (by optical pyrometer) - °F	—	3120	3175	3170	2820	3090.	3100.	
21	Fireroom pressure - Inches of water	4.037	5.937	8.752	9.548	2.739	4.952	5.827	
22	Air pressure in furnace - Inches of water	.372	1.311	3.306	3.707	.16	1.188	1.881	
23	Air pressure at base of stack - Inches of water	-.093	-.046	+.192	+.242	-.115	-.035	+.046	
24	Smoke by Ringelmann scale	0-¼	0-¾	½-¾	½-2	0-¼	0-¼	0-¼	
	FLUE GAS ANALYSIS								
24½	Sampling tube numbers	2-8 19 / 2 / 6	12-18 19 / 2 / 6	12-18 19 / 2 / 6	12-18 19 / 2 / 6	12-18 19 / 2 / 6	12-18 19 / 2 / 6	12-18 19 / 2 / 6	
25	Carbon dioxide - %	13.02 / 12.40 / 11.73	12.27 / 11.90 / 11.93	12.76 / 12.52 / 13.05	13.03 / 12.40 / 13.33	12.90 / 13.45 / 12.30	12.42 / 12.73 / 12.10	12.44 / 11.66 / 12.60	
26	Oxygen - %	3.27 / 3.50 / 4.53	4.72 / 4.45 / 4.87	3.71 / 3.42 / 2.73	3.35 / 3.32 / 2.47	3.53 / 1.75 / 2.26	4.27 / 8.93 / 3.40	4.40 / 4.00 / 2.93	
27	Carbon monoxide - %	.11 / .05 / .04	.02 / .00 / .06	.00 / .12 / .02	.09 / .13 / .00	.04 / .07 / .00	.03 / .00 / .06	.00 / .10 / .03	
28	Nitrogen by difference - %	83.60 / 84.05 / 83.70	83.21 / 83.67 / 83.14	83.53 / 83.93 / 84.27	83.56 / 83.17 / 84.24	83.53 / 84.73 / 84.94	83.30 / 84.34 / 84.45	83.16 / 84.24 / 84.44	
	ULTIMATE ANALYSIS & PHYSICAL TESTS OF FUEL								
29	Carbon - %	86.00	86.00	85.90	85.90	85.90	85.90	85.90	
30	Hydrogen - %	11.70	11.70	11.60	11.60	11.40	11.40	11.40	
31	Sulphur - %	.65	.65	.65	.65	.96	.96	.96	
32	Nitrogen - %	.24	.24	.24	.24	.29	.29	.29	
33	Oxygen and undetermined - %	1.41	1.4	1.6	1.6	1.45	1.45	1.45	
34	Gravity - Baumé	26.1°	26.1°	25.55°	25.55°	25.55°	25.55°	25.55°	
35	Gravity - Specific	.8979	.8979	.900	.900	.900	.900	.900	
36	Flash point - °F	210	210	195	195	215	215	215	
37a	Viscosity temperature - °F.	120	120	120	120	120	120	120	
b	Viscosity - Engler	3.16	3.16	3.06	3.06	3.17	3.17	3.17	
38	Water & sediment - %	.3%	.3%	.3%	.3%	.2%	.2%	.2%	
39	Calorific value - B.T.U. per lb.	19300.	19300.	19300	19300	19280.	19280.	19280.	
	STEAM RESULTS								
40	Superheat - °F.	46.3	62.9	78.7	80.8	39.3	67.4	69.8	
41	Quality of steam leaving drum - %	99.28	99.23	98.60	98.29	99.86	—	—	
42	Factor of evaporation	1.1056	1.1161	1.1222	1.1206	1.104	1.1204	1.1187	
43	Actual evaporation per lb. of fuel - Lbs	14.848	14.261	13.630	13.515	14.978	14.430	14.169	
44	Equivalent evaporation per lb. of fuel - Lbs.	16.416	15.917	15.296	15.145	16.537	16.167	15.851	
45	Actual evaporation per sq. ft. of water H.S. - Lbs. per hour	7.313	10.526	18.434	20.286	5.621	12.189	14.628	
46	Equivalent evaporation per sq. ft. of water H.S. - Lbs. per hour	8.085	11.749	20.687	22.732	6.206	13.657	13.364	
47	% of total heat transfer taking place in superheaters	3.70%	4.57%	5.62%	5.81%	3.34%	—	—	
	OIL RESULTS								
48	Oil burned per sq. ft. of water H.S. - Lbs. per hour	.493	.738	1.352	1.501	.375	.845	1.032	
49	Oil burned per cu. ft. of furnace volume - Lbs. per hour	4.961	7.435	13.623	15.120	3.780	8.509	10.399	
50	Oil burned per burner per hour - Lbs	931.5	930.7	930.1	1032.3	709.8	7.0.0	710.0	
	AIR RESULTS								
51	Relative Humidity - %	47.0	47.3	56.8	52.1	29.4	21.5	21.5	
52	Dew Point - °F.	65.8	65.7	69.5	70.8	65.9	54.1	53.5	
53a	Moisture - Lbs. per cu. ft	.001003	.001000	.001150	.001178	.001006	.000673	.000665	
b	Vapor pressure (e) - Inches of mercury	.634	.632	.720	.752	.636	.419	.410	
54	Weight of dry air - Lbs per cu. ft	.071993	.072629	.073239	.071828	.070337	.07099	.071151	
55	Weight of air at atmospheric conditions - Lbs. per cu. ft.	.072996	.073629	.074389	.073006	.071343	.071669	.071816	
56	Air at atmospheric conditions theoretically required for combustion of one lb. of fuel - Cu. ft.	193.5	191.8	189.4	193.2	196.6	194.7	194.3	
	RESULTS FROM ORSATT & ULTIMATE ANALYSIS								
57	Weight of dry gases per lb. of fuel - Lbs.	16.49	17.85	16.94	16.49	16.71	17.34	17.36	
58	Weight of dry air per lb. of fuel - Lbs.	16.59	17.94	17.04	16.59	16.80	17.41	17.40	
59	Excess air from Orsatt - %	17.6	27.6	20.5	18.0	19.3	24.4	25.4	
60	Volume of air at atmospheric conditions used per lb. of fuel - Cu. ft.	—	—	—	—	—	—	—	
61	Volume of air at fireroom conditions used per lb. of fuel - Cu. ft.	230.5	247.0	232.7	230.8	233.9	245.2	244.6	
62	Velocity of air through vanes of air registers - Ft. per minute	4294	4598.	4329.	4765.	35[illegible]	3498.	3473.	
63	% excess air over theoretical amount required for complete combustion	19.1	28.3	22.9	19.5	21.5	25.9	25.9	
	HEAT BALANCE (BTU / %)								BTU. / %
64	Heat absorbed by boiler	15930 / 82.54	15440 / 80.03	14843 / 76.91	14697 / 76.15	16048 / 83.24	15689 / 81.37	15388 / 79.78	
65	Loss by imperfect combustion	73 / .38	14 / .07	0 / .00	60 / .3	27 / .14	21 / .11	0 / .00	
66	Loss with dry chimney gases	1703 / 8.82	2205 / 11.43	2509 / 13.00	2453 / 12.71	1504 / 7.80	2037 / 10.93	2222 / 11.53	
67a	Loss with moisture from hydrogen in fuel	1296 / 6.72	1339 / 6.94	1379 / 7.15	1374 / 7.12	1229 / 6.37	1290 / 6.69	1300 / 6.77	
b	Loss with moisture in fuel	— / —	— / —	— / —	— / —	— / —	— / —	— / —	
c	Loss with moisture in air	47 / .24	60 / .31	76 / .39	79 / .41	42 / .22	39 / .22	41 / .21	
68	Radiation and error	251 / 1.30	236 / 1.22	493 / 2.55	637 / 3.30	430 / 2.23	154 / .80	329 / 1.71	
69	Total	19300. / 100.00	19300 / 100.00	19300. / 100.00	19300 / 100.00	19280 / 100.00	19280 / 100.00	19280 / 100.00	
70	Boiler and furnace efficiency - %	82.54	80.03	76.91	76.15	83.24	81.37	79.78	

Remarks * Readings used in calculations.

To illustrate paper on "Recent Advance in Oil Burning," by Ernest H. Peabody, Esq., Member.

Table II

FUEL OIL TESTING PLANT

SUMMARY OF TESTS ON NORMAND BOILER.

PEABODY-FISHER BURNERS

Test Number					
Date	8-26-20	8-25-20			
Barometer-in. of mercury	- - -	- - -			
Duration of Test-hours	2	2			
Number of Burners	3	3			
Type of Burners	PEABODY FISHER	PEABODY FISHER			
Type of Air Registers.	Peabody	Peabody			
Steam Pressure on Drum -lbs. /sq.in.g.	240	240			
Steam Pressure below Stop -lbs. /sq.in.g.	- - -	- - -			
Superheater Temperature - °F.	- - -	- - -			
Feed Water Temperature - °F.	204.0	202.2			
Water Evaporated -lbs./hr.	29000	65457			
Oil Burned -lbs./hr	2048.5	5408			
Oil Pressure at Burners -lbs./sq.in.g.	203.2	273.6			
Oil Pressure on Return Line -lbs./sq.in.g.	Closed	Closed			
Oil Temperature at Burners - °F.	256.4	260.4			
Fireroom Temperature - °F.	94.5	105			
Stack Temperature - °F.	675	928			
Fireroom Pressure - in. of water	‡‡1.86	8.3			
Smoke (Ringelmann Scale)	½ - ¾	1 - 1½			
Flue Gas - CO_2 - %	12.27	15.2			
Flue Gas - O_2 - %	5.1	0.5			
Flue Gas - CO - %	0	0			
Calorific Value of Fuel - B.T.U./lb	18129	18129			
Superheat - °F.	- - -	- - -			
Factor of Evaporation.	1.064	1.063			
Actual Evaporation per lb. of oil-lb	14.16	12.10			
Equiv. Evaporation per lb. of oil-lb.	15.07	12.86			
Oil Burned per Sq.Ft. H.S.-lbs./hr	.455	1.202			
Oil Burned per CO.FT.F.V.-lbs/hr	4.21	11.15			
Oil Burned per Burner per hr.-lbs.	682.8	1802.7			
Boiler and Furnace Effeciency - %	80.67	68.84			
Remarks.	‡‡ Draft in Uptake.				

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Table III

SUMMARY OF TESTS ON WHITE-FORSTER BOILER

Test Number	44-8	44-6	44-11	44-7	44-12	******	
Date	9-29-20	9-15-20	9-30-20	9-16-20	10-1-20	9-17-20	9-17-20
Borometer-in. of mercury	------	30.056	--------	29.94	-----	30.10	30.10
Duration of Test-hours.	3	3	3	3	1	1	½
Number of Burners.	3	3	5	5	6	3	3
Type of Burners.	B&W	B&W	B&W	B&W	B&W	B&W	B&W
Type of Air Registers	"Lodi"	"Lodi"	"Lodi"	"Lodi"	"Lodi"	"Lodi"	"Lodi"
Steam Pressure on Drum-lbs.sq.in.g.	284.9	276.8	281.3	281.1	284.7	281.3	285.8
Steam Press.belowStop-lbs.sq.in.g.	279.8	268.4	269.9	261.4	255.9	267.2	272.8
Superheater Temperature-oF.	462.4	470.7	490.5	494.1	493.1	487.8	487.4
Feed Water Temperature-oF.	200.1	200.3	200.0	200.8	200.0	200.5	200.4
Water Evaporated-lbs. hr.	43978	63695	83890	102222	119065	86625	89270
Oil Burned-lbs. hr.	3054.4	4552.6	6069.6	7580.6	8989	6714	6862
Oil Press.at Burners-lbs.sq.in.g.	204.3	257.8	266.5	269.8	261.3	324.6	341.4
Oil Press.on Return Line " " "	-----	-----	-----	-----	-----	-----	-----
Oil Temperature at Burners-oF.	155.9	146.4	157.6	147.2	157.0	147.2	146.3
Fireroom Temperature-oF.	102	105.9	-----	103.8	-----	100.9	-----
Stack Temperature-oF.	567	632	675	714	780	667	709
Fireroom Pressure-in.of water	2.87	4.44	5.57	6.60	7.51	6.94	7.12
Smoke (Ringelmann Scale).	0-¼	0-¼	0-¼	0-¼	0-¼	½	½
Flue Gas-Inner Pass-CO2-%	13	12.89	13.1	12.2	14.0	11.3	12.8
Flue Gas-Inner Pass-O2-%	3.3	3.15	2.8	3.83	1.9	5.3	3.0
Flue Gas-Inner Pass-CO-%	0.0	0.0	0.0	0.0	0.0	0.0	0.0
Calorific Value of Fuel-B.T.U.,lb.	18939	18988	18994	18973	18994	18939	18939
Superheat-oF.	48.22	58.5	79.4	83.9	87.0	76.0	75.8
Factor of Evaporation	1.1018	1.1083	1.1267	1.1221	1.1239	1.1182	1.1173
Actual Evaporation per lb.of oil-lbs.	14.399	13.991	13.821	13.485	13.246	12.902	13.009
Equiv.Evaporationper lb.of oil-lbs.	15.855	15.506	15.459	15.132	14.887	14.427	14.535
Oil Burned per Sq.ft.H.S.-lbs.hr.	.404	.602	.802	1.002	1.1882	.888	.908
Oil Burned per Cu.ft.F.V.-lbs.hr.	4.067	6.06	8.082	10.09	11.969	8.940	9.138
Oil Burned per Burner per Hr.-lbs.	1009.1	1517.5	1213.9	1516.1	1498.2	2238	2287.3
Boiler and Furnace Efficiency-%	81.29	79.24	78.98	77.39	76.05	73.92	74.47
Remarks							

*****The tests of 9-17-20 are unofficial because of three tubes in boiler being plugged.

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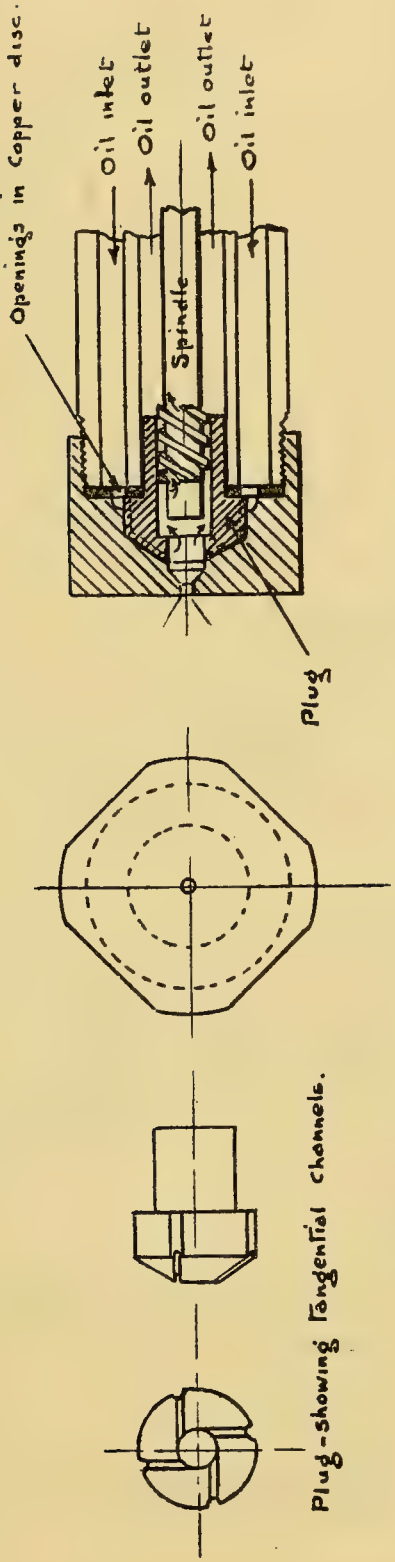

FISHER BURNER USED IN U. S. NAVY

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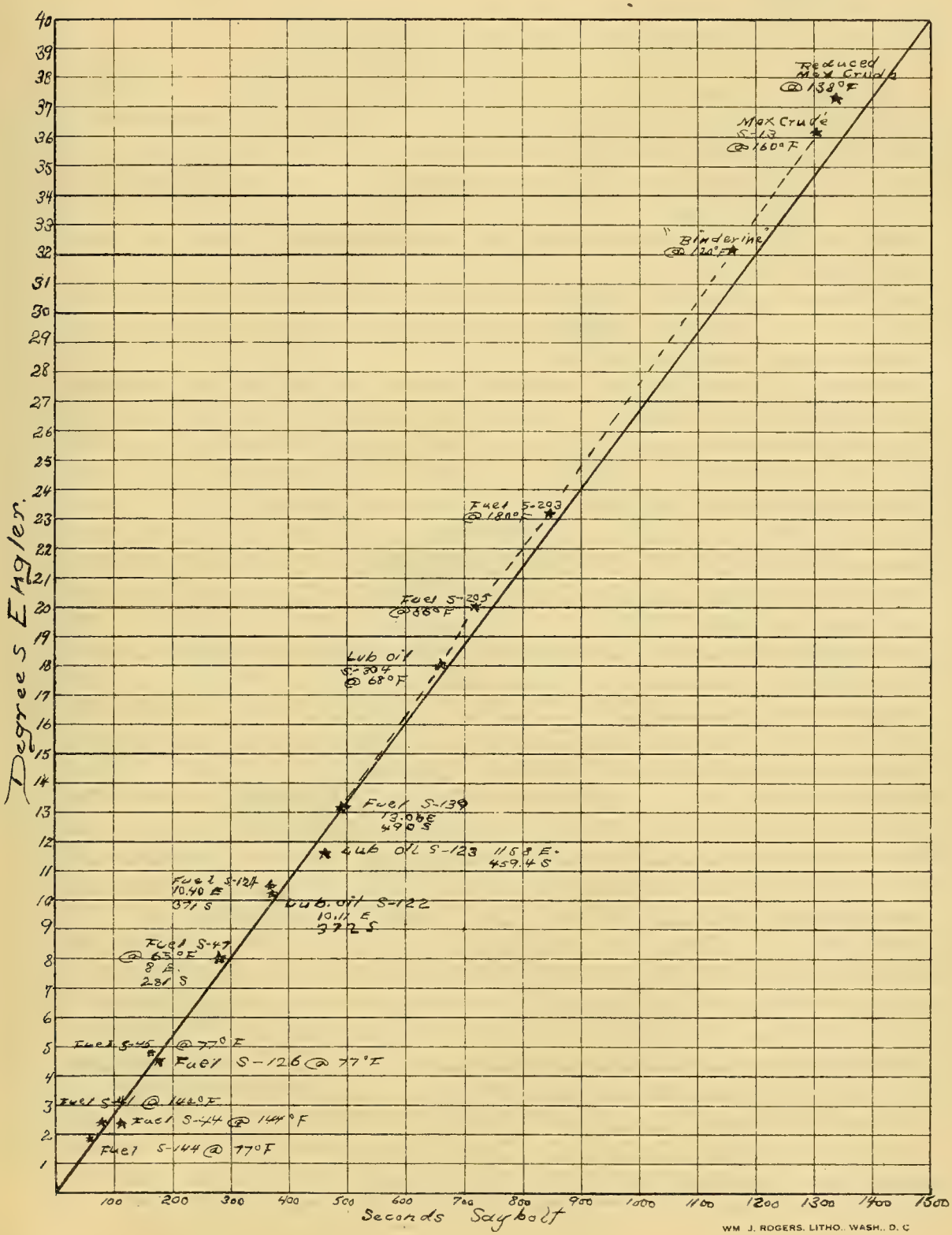

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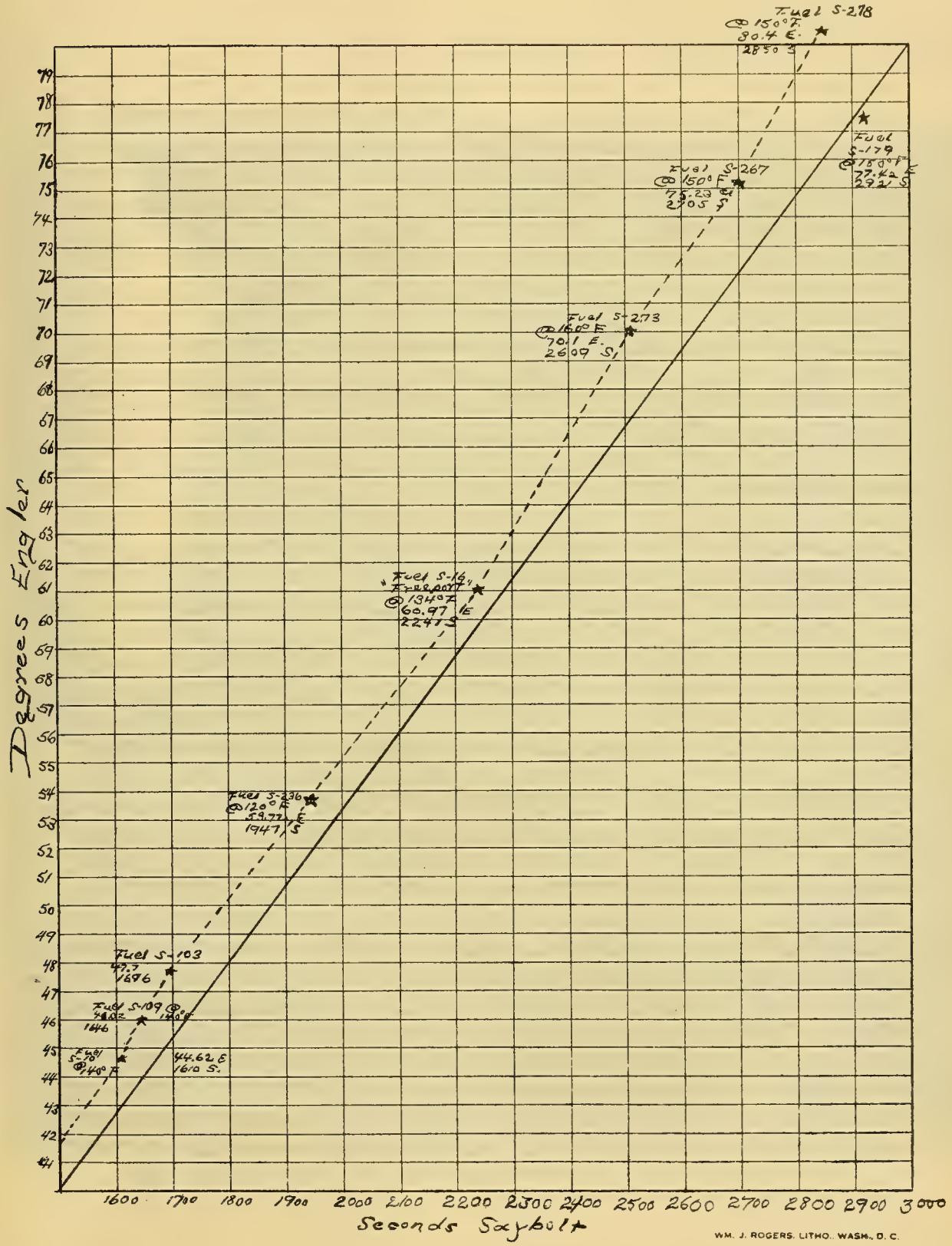

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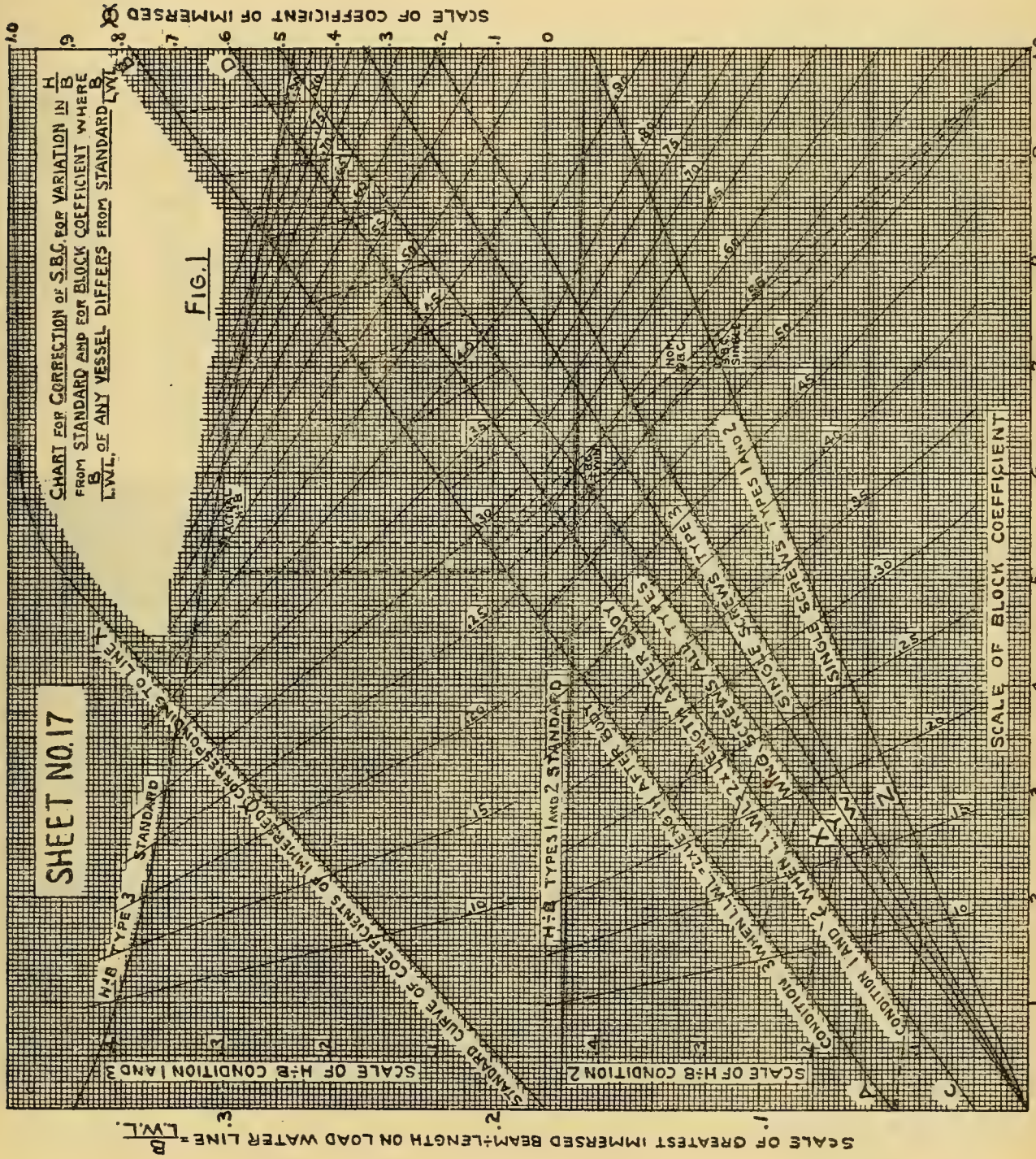

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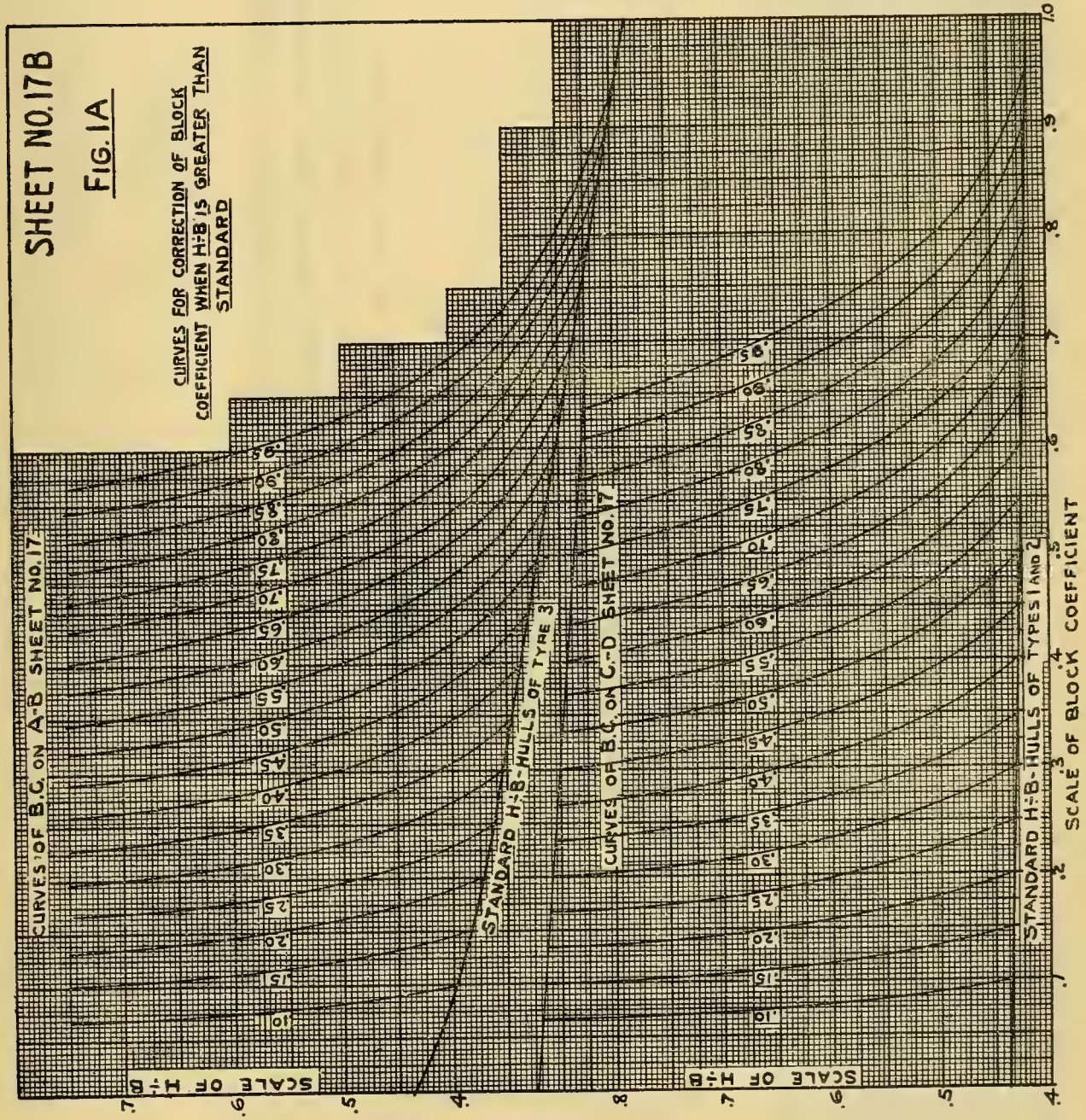

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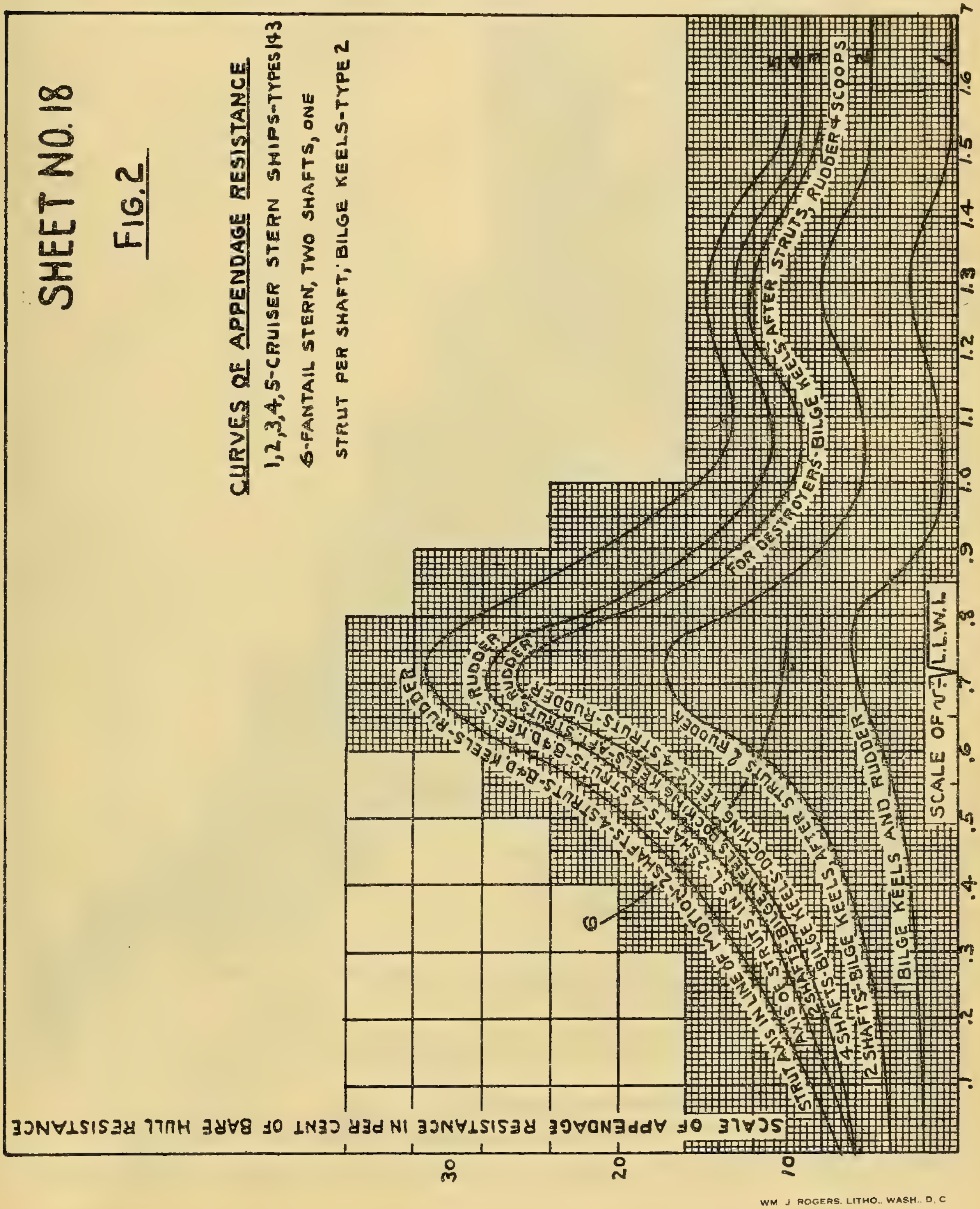

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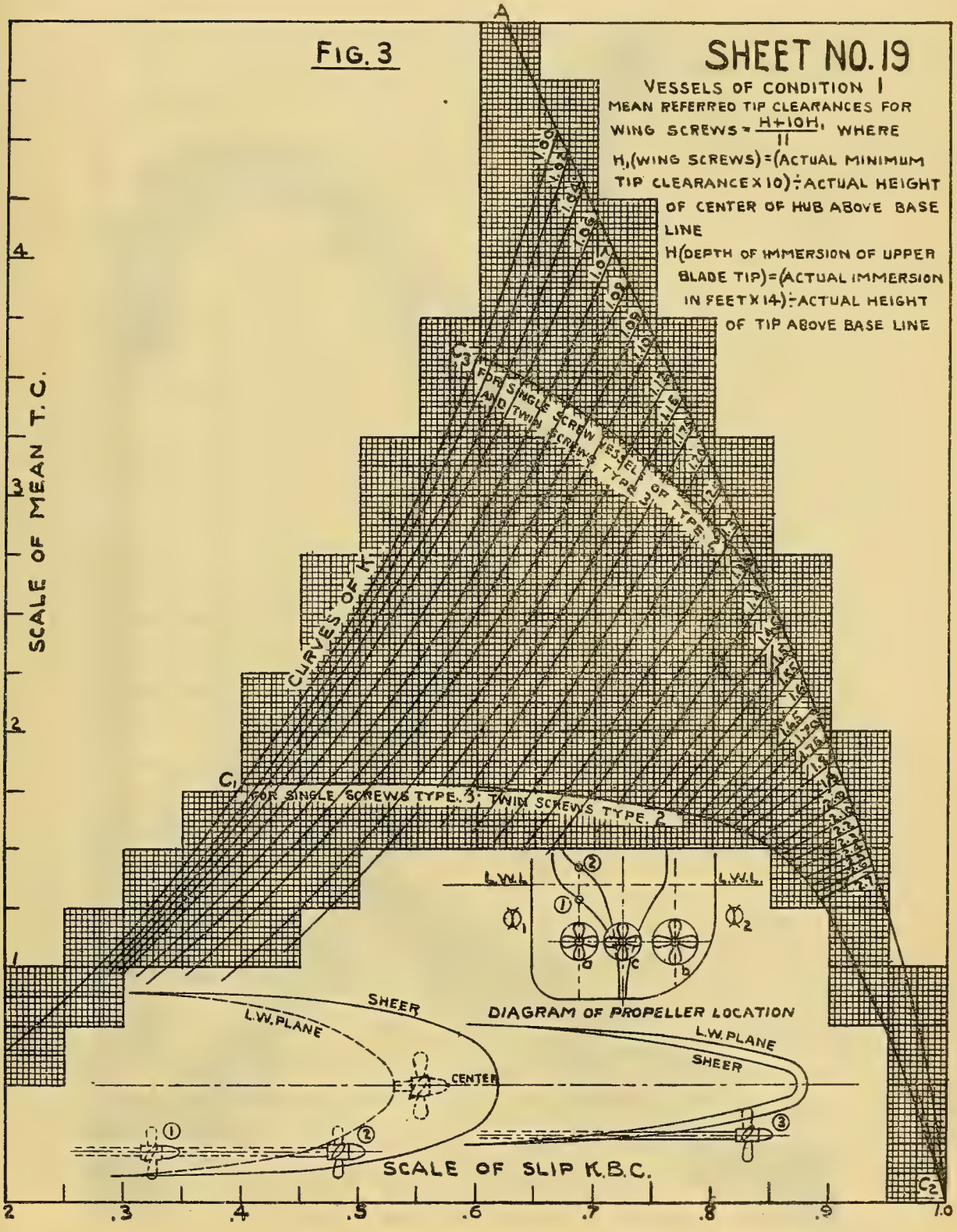

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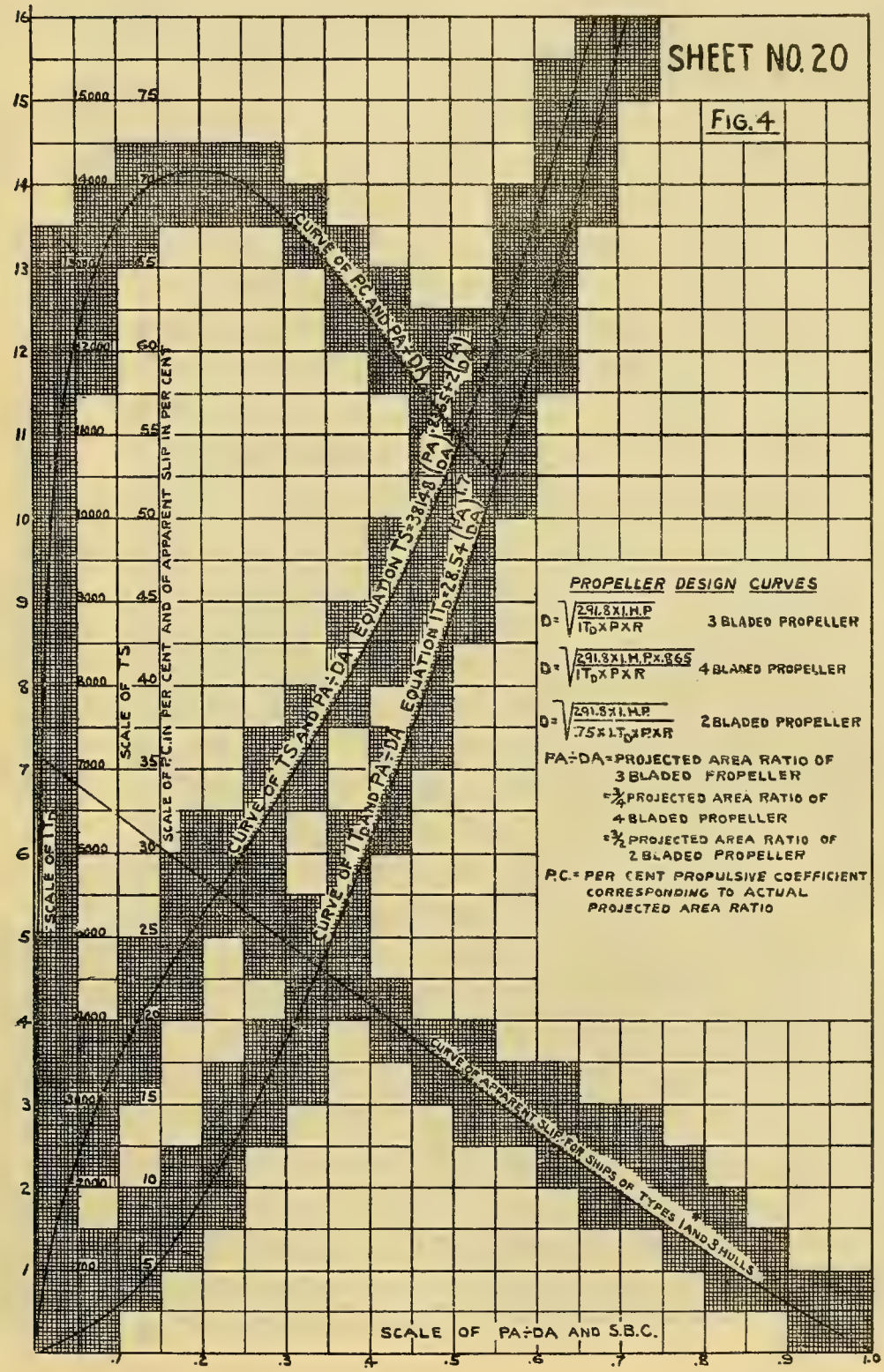

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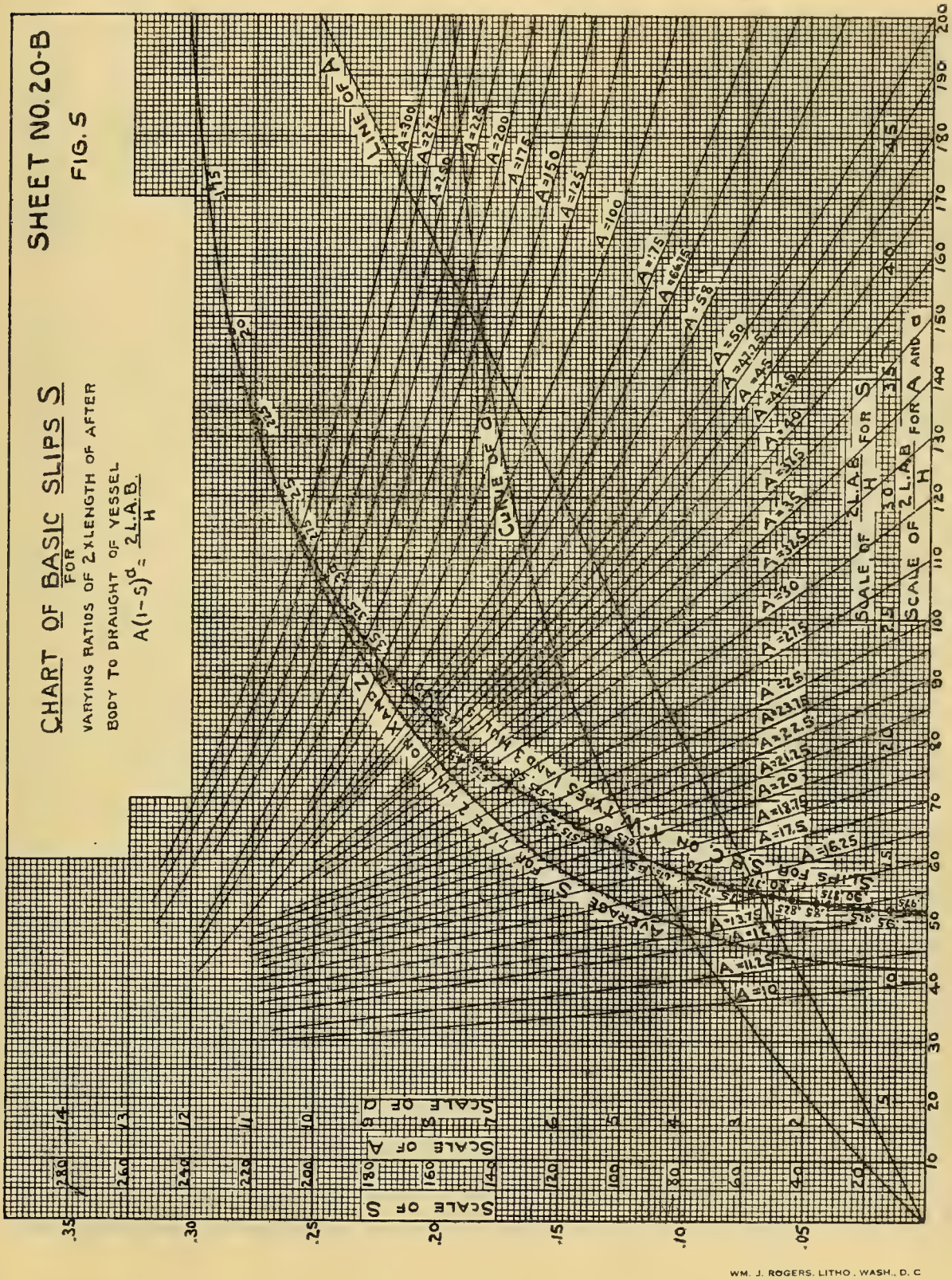

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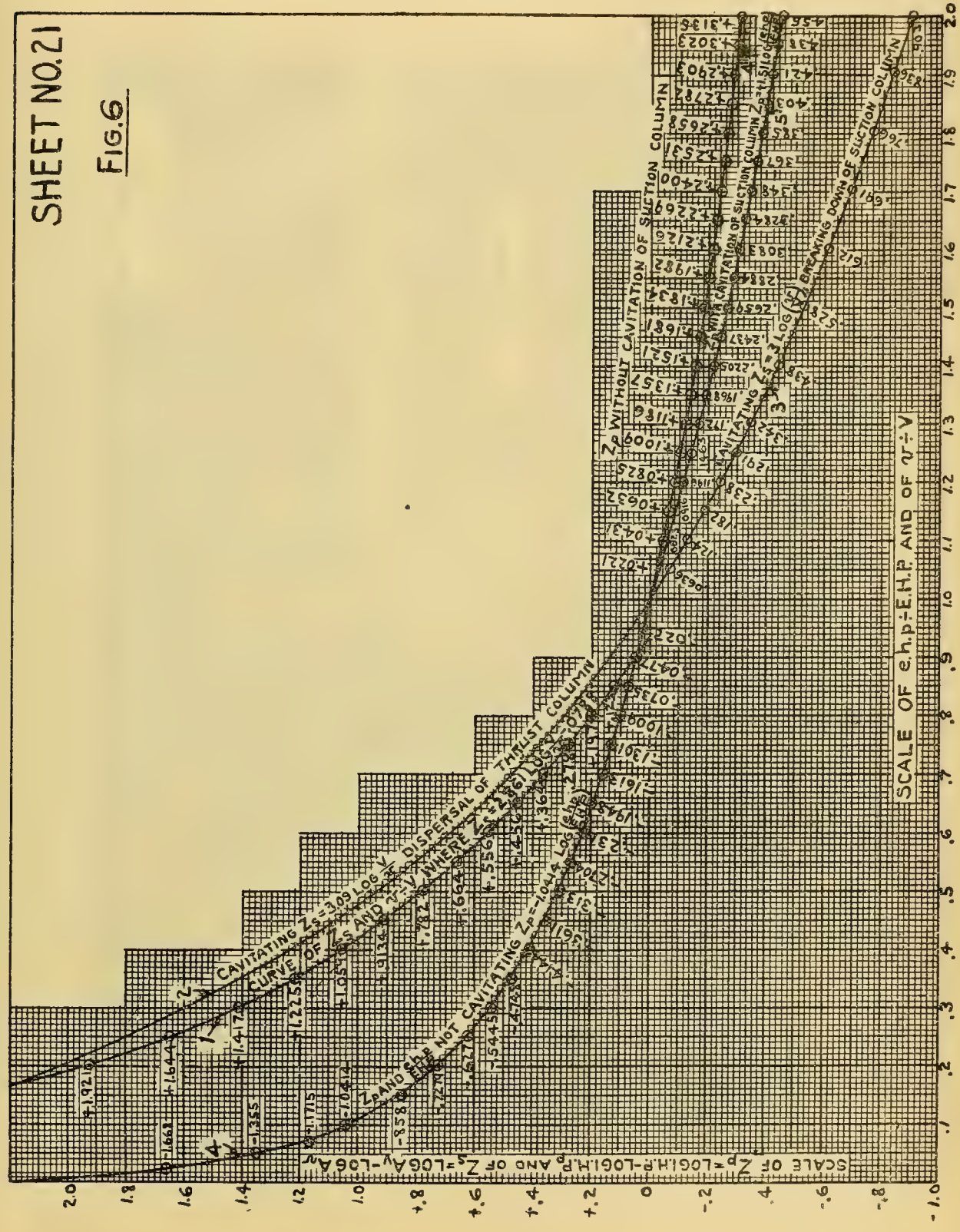

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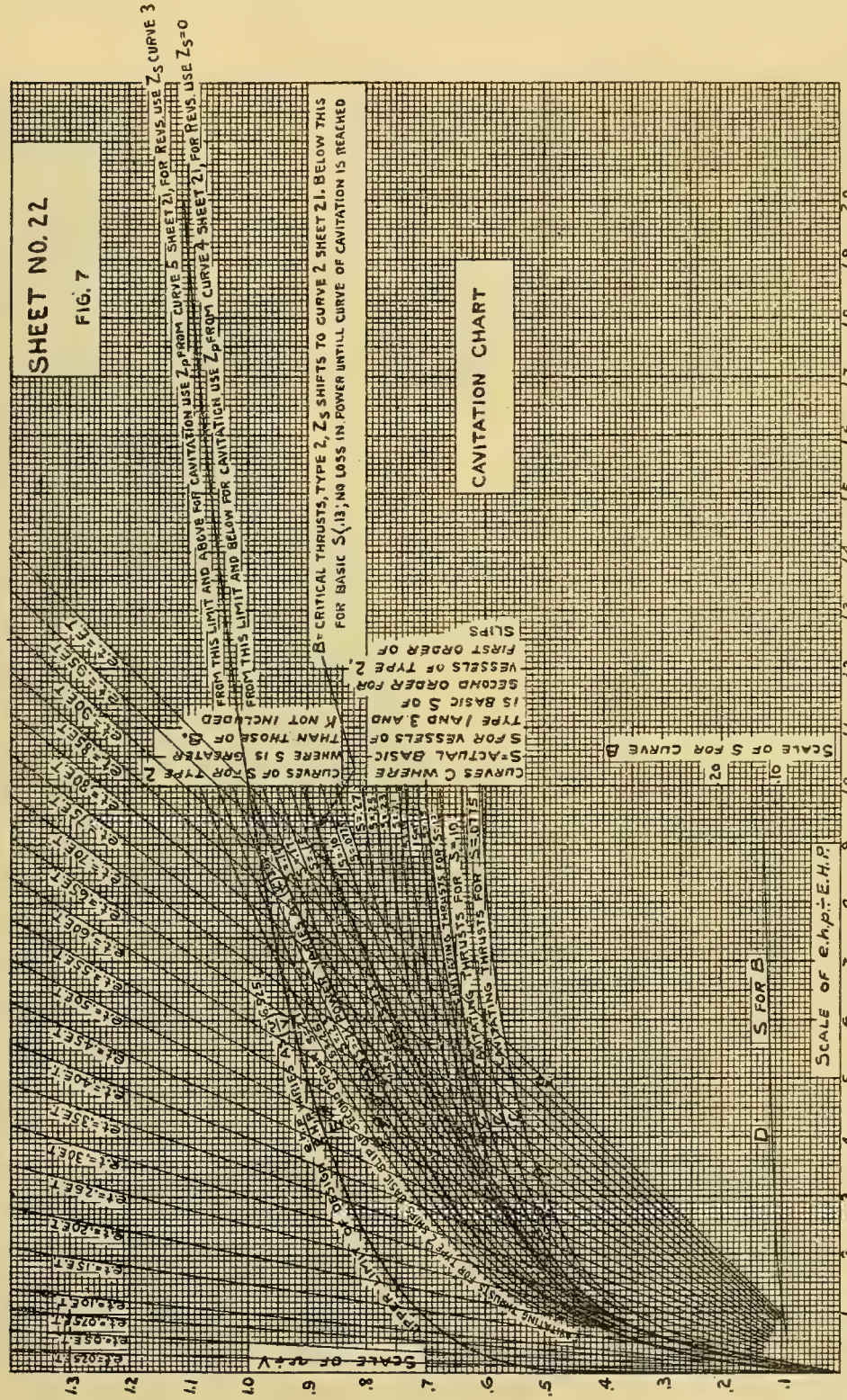

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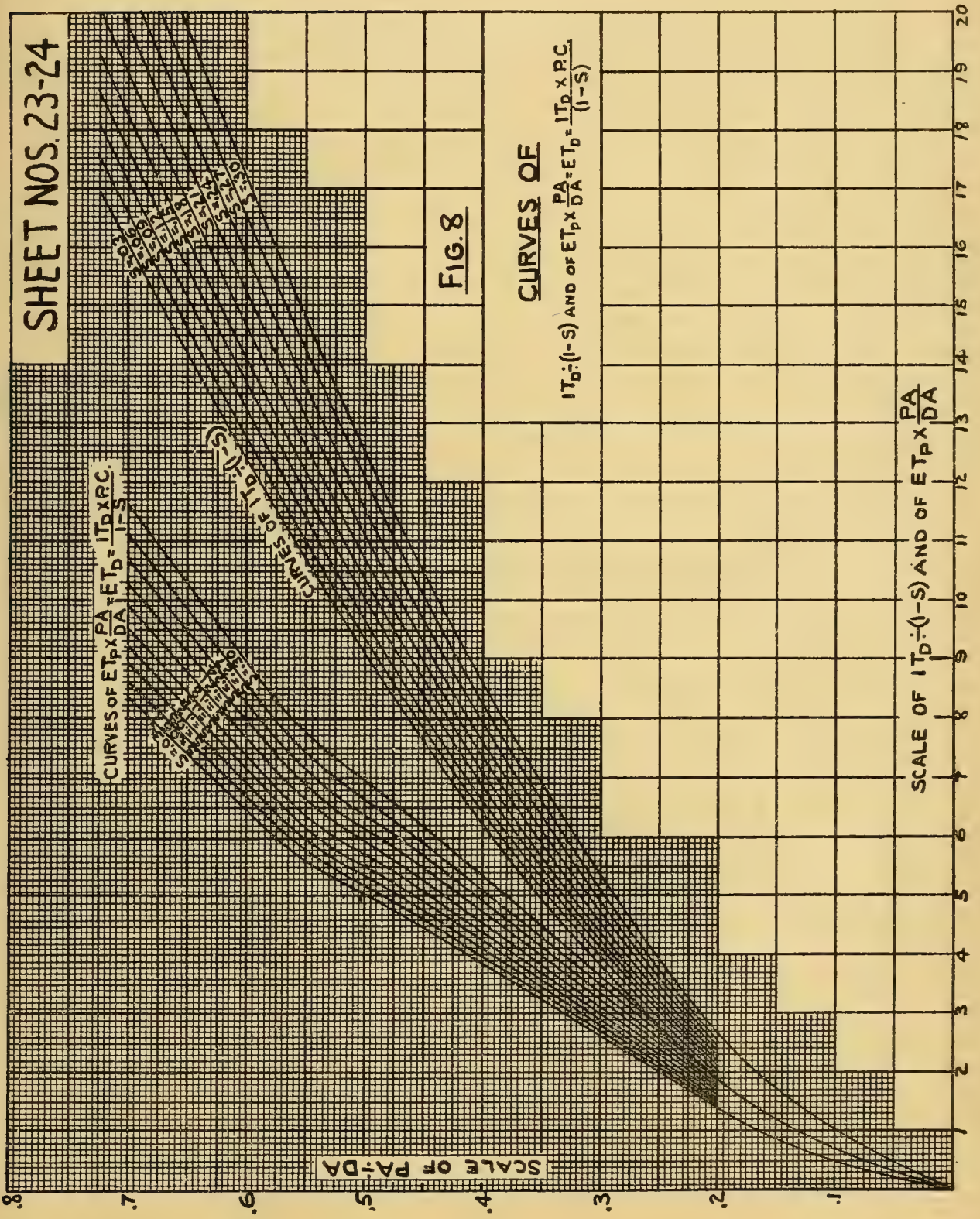

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